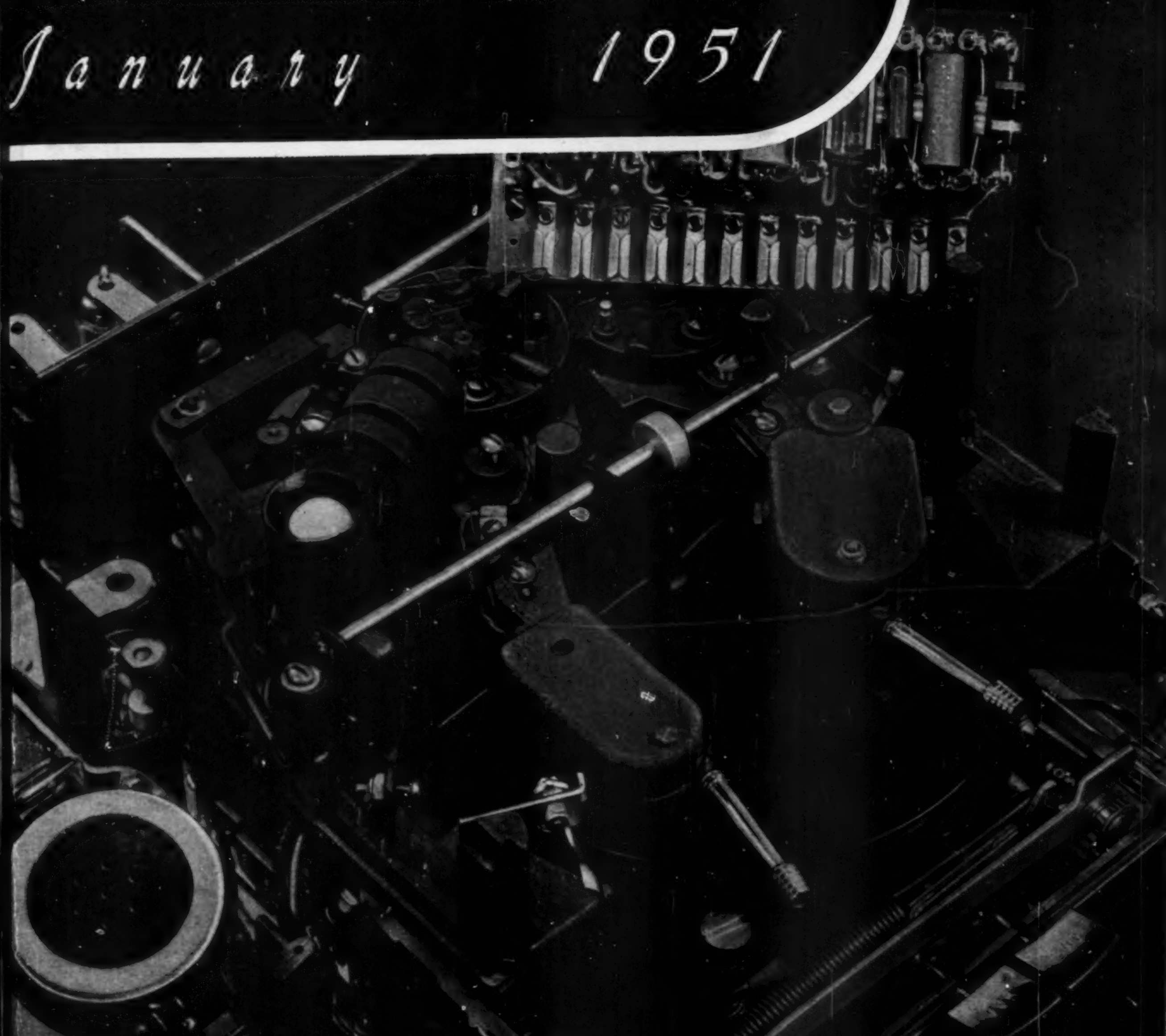
MACHINE DESIGN

January 1951



B-13

MAIN OFFICE

Penton Building, Cleveland 13, Ohio
Main 1-8260

Branch Offices

New York 17 ...... 60 East 42nd St. Murray Hill 2-2581
Chicago 11 ...... 520 North Michigan Ave. Whitehall 4-1234
Pittsburgh 19 ...... 2837 Koppers Building Atlantic 1-3211
Detroit 2 ...... 6560 Cass Ave. Trinity 5-3024
Washington 4 ...... 1123 National Press Bldg. Executive 6849
Los Angeles 4 ...... 130 N. New Hampshire Ave. Dunkirk 2-1758
London, S.W.1 .. 2 Caxton St., Westminster

Published by
THE PENTON PUBLISHING COMPANY

E. L. SHANER ...... *Chairman*
G. O. HAYS ...... *President and Treasurer*
R. C. JAENKE ... *Vice Pres., Director of Adv.*
F. G. STEINEBACH .. *Vice Pres. and Secretary*
R. W. COLEMAN ...... *Asst. Treasurer*

Also publisher of STEEL • THE FOUNDRY
NEW EQUIPMENT DIGEST

Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15.

Acceptance under Act of June 5, 1934. Authorized July 20, 1934.

MACHINE DESIGN

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

Vol. 23—No. 1 **January, 1951**

DESIGN FOR PRODUCTION • STYLING • MATERIALS SPECIFICATION • DESIGN ANALYSIS • MACHINE COMPONENTS • ENGINEERING MANAGEMENT

Over the Board

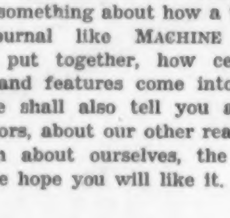
Introducing Ourselves

In the course of our travels each of us on MACHINE DESIGN'S editorial staff visits annually not more than a few hundred of you, our 66,000 readers. With most of you, therefore, we have no opportunity to get acquainted, to talk about your problems and ours, to swap stories—in short, to chew the rag over the board.

To help overcome this lack of personal contact we are introducing this page of informal comment. Here each month we shall try to give you glimpses into our editorial workshop, tell you something about how a technical journal like MACHINE DESIGN is put together, how certain articles and features come into being. We shall also tell you about our authors, about our other readers, and even about ourselves, the editors. We hope you will like it.

This Month's Cover

Did you ever look inside a modern dictating machine? One time when our own office equipment was being serviced we were amazed at the intricate and colorful components nested in a compact array inside a magnesium housing. To show you what we saw, Jack H. Stock made the color photograph which is reproduced on the cover of this issue. Naturally, it became the subject of the article beginning on Page 90. Although not all MACHINE DESIGN articles take so long to develop, it is noteworthy that almost a year of conferences with the author, gathering additional data, taking more pictures, and revising the manuscript were required before we felt the article told the story you would want.



Return Postcards

We haven't checked this, but we would be surprised if MACHINE DESIGN doesn't hold some kind of a record for the number of return postcards included in each issue (see opposite Pages 32, 152 and 168). The November and December issues included a single card on which you could request extra copies of articles in the editorial section. Your response has indicated a real appreciation of this service, and to meet the needs of our multiple readership we are providing three cards in this issue (opposite Page 152). We are printing enough "overruns" of the editorial section to meet expected demands. If we start running out of stock because of your requests, we'll print more in the future.

New Parts and Materials

For some time we have felt the need for announcing new parts and materials more effectively — in a manner more directly useful to you. The outcome of our thoughts along this line is now in the third month of its trial run, and begins on Page 157 of this issue. Use of a datasheet approach giving parallel design information on all standard machine components and materials will, we hope, prove to be the best answer to the designer's need for a quick, accurate reference. Then, too, these small data sheets can be pasted on cards to create a rather complete, self-indexing file on standard components and materials.

In developing this new presentation we have become extremely conscious of the shortcomings of the conventional approach—the missing data and the ambiguities. Considerable followup has been necessary to insure completeness and accuracy in each item. If you like this novel approach—or if you think it could be improved—why not drop us a line?

Strain Gages

Strain gages are usually thought of primarily as elements for measuring strain in stressed machine elements. However, when we found that Sergei Guins of the C. & O. railway's research department was obtaining directly applicable design information by using strain gages to measure deflections and movements on operating railroad equipment we felt sure that here was a really practical innovation with great potentialities. Obtaining measurements of any kind aboard a moving train is vastly different from the same job in a nice clean, quiet lab. Our hats are off to Sergei for his perseverance, the results of which are reported for your benefit in his article beginning on Page 145.

A Personal Problem

As part of our continuing study of the major problems of design engineers we've been asking you a lot of questions via the U.S. mails. A recent questionnaire listed several subjects such as cost reduction, materials substitution, better appearance, reduced weight, and the like, with a request that you check those items of greatest immediate concern to you. One reader checked weight reduction, which he said was purely a personal problem!


PIONEER MANUFACTURER
OF QUALITY
"O" RING SEALS
THE NATION'S LARGEST
EXCLUSIVE PRODUCER
PRECISION RUBBER
PRODUCTS CORPORATION
MAIN OFFICE AND PLANT
3110 OAKRIDGE DRIVE · DAYTON 7, OHIO
BRANCHES IN ALL PRINCIPAL CITIES

ITEMIZED INDEX

Classified for convenience when studying specific design problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

Enduro®
STAINLESS STEEL

THRIFTY METAL OF 10,000 USES

Aeronautical advance—with accompanying fierce heats, high pressures, fantastic speeds—calls for super-metals capable of self defense from several directions at once. Such a metal is ENDURO Stainless Steel—one of aero-engineering's keenest and most useful tools!

...SURVIVES THE TORTURE CHAMBER

● And what a torture chamber—this 1200-pound, J-34 axial turbo-jet! High-temperature gases roar from this inferno at 1,200 m.p.h. It develops a ton-and-a-half jet thrust—and packs 100 tons of air into the compression chamber every hour.

Obviously, only super-metals can survive the ordeal—can stand the gaff and continue to come through. Among them is stainless steel. Here in rotor compression blades, it whirls at 12,500 r.p.m. Here, too, in baffles, shrouding bands and nozzle diaphragm assemblies it fights both terrific heat and tremendous pressure.

That ENDURO Stainless Steel can take it is no surprise. Being an alloy steel, it possesses great strength and ruggedness. Its strength-to-weight ratio is extremely high—its rust- and corrosion-resistance correspondingly great.

BUT—are *you* making *full* use of its capabilities? May we tell you more?

✓ **CHECK ALL 12 ADVANTAGES:** Rust- and Corrosion-Resistance • Heat-Resistance • High Melting Point • Low Coefficient of Expansion • High Strength • Good Dimensional Stability • No Metallic Contamination • Easy to Clean • Easy to Fabricate • Eye Appeal • Long Life • Low End Cost • *What more can be desired in a material?*

REPUBLIC STEEL

For Complete Details Write

REPUBLIC STEEL CORPORATION

Alloy Steel Division, Massillon, Ohio • GENERAL OFFICES, CLEVELAND 1, OHIO • Export Dept.: Chrysler Bldg., New York 17, N.Y.

TOPICS

REVERSIBLE MAGNETIC - CLUTCH transmission has been developed by Vickers Electric Div. Utilizing dry iron and graphite particles suspended in oil, the clutch forms a rigid connection between driving and driven members when current is applied. In the reversible transmission two separate electromagnetic coils control the action of two intermediate gears. Exciting one coil produces forward motion of the output shaft; the other, reverse motion. Different speeds in the two directions are possible. Also, the system can be designed to give two different speeds in the same direction.

HIGH PRESSURE intensifiers, pipe and fittings capable of handling up to 200,000 psi have been developed by the Harwood Engineering Company. By a modification of the autofrettage principle used in ordnance manufacture, compressive stresses are introduced to counteract high tensile stresses developed in components of high-pressure equipment. For added strength and toughness, heat-treated double-tempered alloy steels of the nickel-chromium and nickel-chromium-molybdenum types are used.

PROCESS FOR BRIGHTENING ALUMINUM and its alloys with a low-concentration acid bath was recently announced by the Kaiser Aluminum & Chemical Corporation. The process is finding application in brightening mill-finish sheet, brightening articles for anodizing and coloring, increasing total reflectivity of buffed articles, and brightening articles without buffing.

SIMULATED CORROSION TESTS have no correlation with actual corrosion found under any circumstances. According to F. L. LaQue of the Inco Harbor Island Corrosion Testing Station, there is also no indication of relative merit among various metals concerning their corrosion resistance. Tests over the last 15 years show that actual environmental conditions only will give useful results.

NEW REGULATOR, known as a microsyn, provides regulation in infinitesimal steps. Basic idea for the device originated at MIT and developments were extended by the Westinghouse Electric Corporation. A microsyn can produce a voltage proportional to rotation of a rotor over a range of plus or minus ten degrees with an angular sensitivity of 20 seconds, can produce torque that is linear with the square of signal current, or it can act as an elastic restraint generator, producing torque proportional to the product of angular displacement and the square of current.

COOLANT EFFICIENCY is increased by the addition of minute quantities of a new wetting agent developed by the Aquadyne Corporation. Known as Hydrodyne, it utilizes the basic principle of "wet water" in reducing the interfacial tension of the coolant. It permits the coolant to make intimate contact with working surfaces and to dissipate heat more rapidly.

800,000 REVOLUTIONS PER SECOND is the speed attained with steel rotors of about 0.01-inch diameter on centrifuges with magnetic suspension.

FIRST ALUMINUM SHIP of all-welded construction was launched recently by the Navy. First of a group of four in a new 95-foot PT class, the ship has a beam of 25 feet, a standard displacement of 75 tons, and four torpedo tubes.

STAINLESS STEEL rolled at −300 F exhibits significant improvement in physical properties according to a report of co-operative studies conducted by Crane Company and Westinghouse Electric Corporation. A combination of preparatory heat treatment, subzero working, and subsequent high temperature aging was found to give increased hardness and strength. Proportional limit was doubled; also, torsional yield stress and fatigue strength were increased by about one half.

GREASES with an effective lubricating range of −65 C to 250 C have been developed by the Naval Research Laboratory in co-operation with industrial organizations. Copper phthalocyanine, a brilliant blue organic chemical used to gel the lubricant fluids, is the key to the new greases. Service at 250 C is possible because of a fluid silicone base. Lifetime lubrication of electric motors and extended shelf life of equipment are expected advantages.

Revere Brass of carefully controlled grain sizes is used in the Andirons, the Fire Lighter, the Lighter Cover and the Torch Handle.

• Sometimes a routine laboratory procedure finds ways to make improvements even when everything already is "completely satisfactory". In fact, that is one of the main reasons for carrying out laboratory routines.

A case in point is the Decorative Polished Brass Fire Lighter produced by Peerless Manufacturing Corp., Louisville, Kentucky.

Here is a product that was rolling down the production line and on into homes all over the country. The consumers were satisfied and Peerless was pleased with the appearance of its product. There were no troubles. Nevertheless, the Revere Technical Advisory Service was asked to study the polishing methods and find out if even better procedures would be advantageous.

Just as a routine procedure our laboratory men cut up several of the partly drawn "Pots" and checked on the gauge diminution caused by drawing. The "Lab. Men" are continually doing things like that . . . studying the successful products in order to pile up data which may be useful when they run into a "problem" product.

They found that with a different drawing sequence the draws, although still deep, could be made less severe. The new drawing sequence would permit the use of smaller grained metal. The smaller grain would make polishing easier, even though the product as it went out into the market could have *no* more than the same highly polished beauty it always had.

By testing to find if it could get one cost saving, this company got two.

Perhaps you also are thinking in terms of one slight improvement when two or more are readily available. The Revere Technical Advisory Service offers the laboratory routines which will find out. If you use copper, brass, bronze, aluminum, nickel silver—any alloy which Revere can make—just get in touch with the nearest Revere sales office.

REVERE

COPPER AND BRASS INCORPORATED

Founded by Paul Revere in 1801

230 Park Avenue, New York 17, New York

• • •

Mills: Baltimore, Md.; Chicago and Clinton, Ill.; Detroit, Mich.; Los Angeles and Riverside, Calif.; New Bedford, Mass.; Rome, N. Y.
Sales Offices in Principal Cities, Distributors Everywhere.

JANUARY, 1951

MACHINE DESIGN
A PENTON PUBLICATION

To Engineers—An Opportunity and a Challenge

INCREASING concern is being voiced today over the scarcity of men competent to assume executive responsibility in industry. Why? Because the complexity of modern industrial operations demands managers with a high level of technical ability and experience who are also trained to cope with economic and human problems.

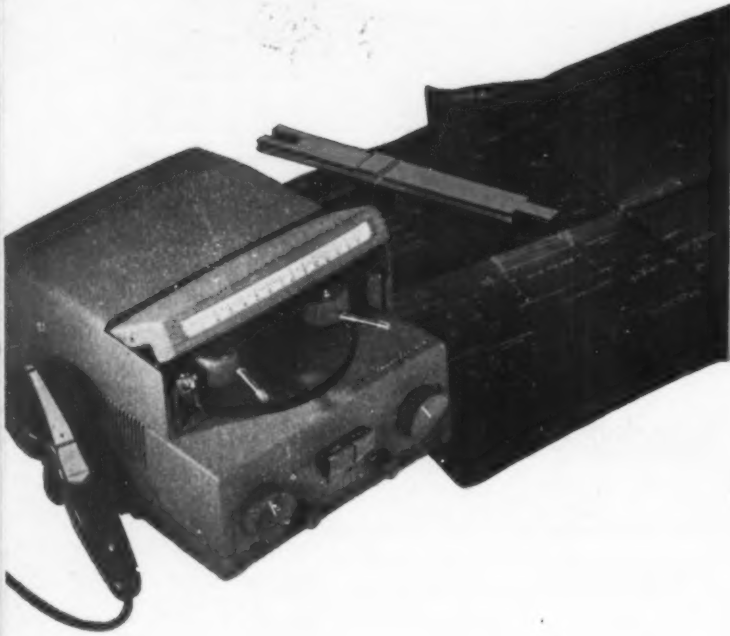
Who will fill these jobs? There seems to be only one good answer, inasmuch as a man trained solely on the business side—well versed though he may be in the solution of economic and human problems—stands only a slim chance of acquiring a real understanding of technical problems. The final choice must rest on the engineer, in spite of his reputed absorption with technical problems to the virtual exclusion of other aspects of our industrial economy.

By the very nature of their work engineers are specialists. As Bruce Wallace has pointed out, they have concentrated their efforts on 10 to 15 per cent of the sales dollar—and done a marvelous job of making more goods available at lower costs. But many of them need to broaden their scope and realize the tremendous opportunities that exist for them in the control of that other 85 to 90 per cent of the sales dollar—in top management. No group of people in industry is better trained than the engineers to solve complex problems that require careful analysis, complete understanding, unbiased reasoning, and the ability to express their findings in unmistakable language.

How can an engineer hitherto concerned primarily with technical problems broaden his horizons? Embarking on a campaign of self-education, he can look beyond his concentration on a specialized field, begin to learn about the overall operation of business in general and his own company's business in particular—such problems as labor relations, wage incentives, cost control, budget systems, financing, and the essentially human problems of how to keep up production and deliveries in the face of supply and labor difficulties. Complete familiarity with and understanding of these fields plus sound engineering competence and experience add up to an unbeatable combination. With sufficient engineers possessing these assets there should be no difficulty in filling top management jobs with top-flight managers.

Colin Carmichael

EDITOR



DICTATIN

By Lincoln Thompson
Vice President
The SoundScriber Corp.
New Haven, Conn.

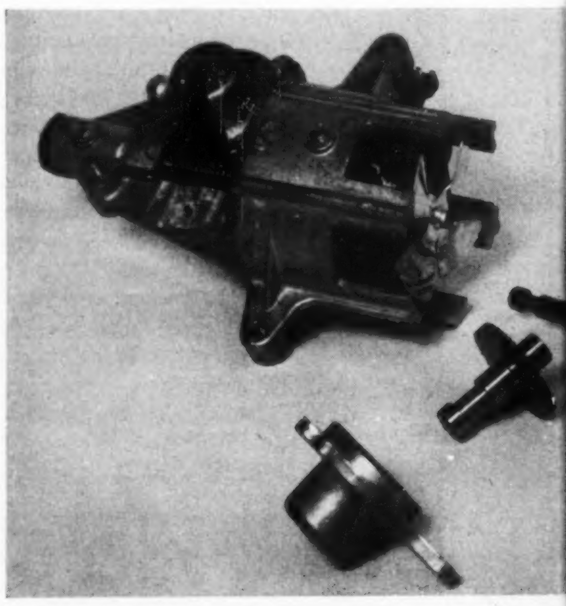
COMPLEXITY of all kinds of transactions and communications have made office costs increasingly larger. Accordingly, management is becoming more aware of the need for increase in efficiency, elimination of waste and reduction of unproductive effort. As a result, greater and greater emphasis is being placed on all methods of time-saving in office procedures, particularly mechanization. Dictation machines are an important contribution both as a means of writing letters and as an executive aid. Saving of executives' time, saving of secretarial time, and providing the ability to put down thoughts as they come are important characteristics of a dictating system.

The designer of dictating machines must have before him at all times the concept that they are the production machines in the office. He must solve many unusual and interesting problems to provide more convenient operation, better clarity and efficiency, increased durability, greater flexibility of use, and more safeguards to produce better results. All of these features are to be contained within attractive appearing enclosures whose bulk and weight must be kept at a minimum.

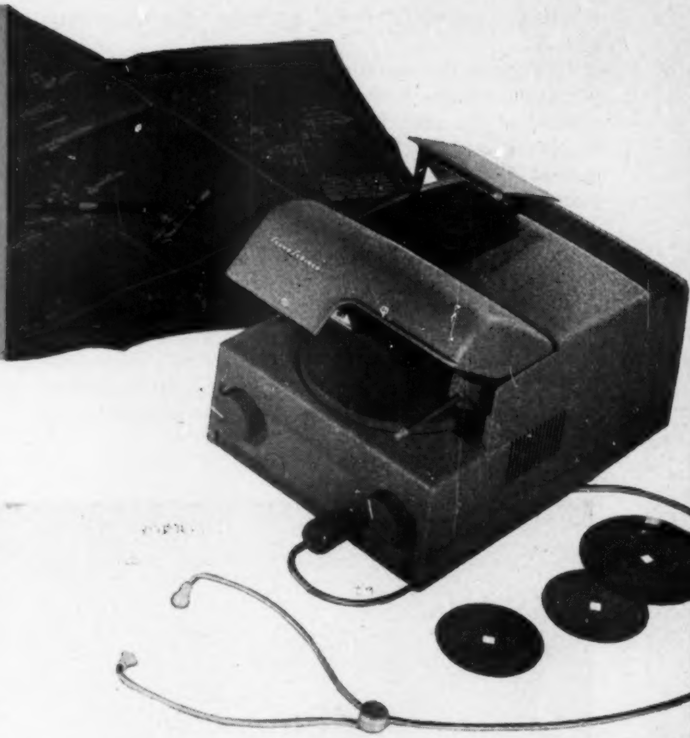
In 1940, SoundScriber Corp. introduced plastic-record instruments which successfully combined electronic amplification and the use of a thin, plastic record with long playing time. In designing the new models for recording and transcribing, shown in the head illustrations, the objective was to provide designs which would fully realize the potentialities of the plastic record and combine them with improvements in electronic, mechanical, optical and recording art. An example of the thoroughness of the approach was the motion analyses which were made of operations such as taking off and putting on the record. Fortunately, the many elements of the overall problem could be separated into different independent projects which could be developed separately and then put together as the research and design progressed. Accordingly, the design background can be stated as a series of problems and their solutions.

The major elements of a dictation recorder are:

1. Mechanical drive system, including mechanisms for instant start and feed
2. Talk and listen-back features, including the microphone
3. Time scales and indexing systems for stylus location and for indication of length of letters, corrections, and special information
4. Electronic system comprising the amplifier con-



MACHINES



... designed for foolproof operation, convenience in use, lightweight and compact construction and pleasing appearance

trols and the recording and playback heads
5. Put-on, take-off and general record handling features
6. Special warnings and safeguards to assure freedom from trouble at the transcribing end with a minimum of attention by the dictator.

These elements find their counterparts in the transcriber, which is the end point of the dictating operation, and include:

1. Mechanical drive system, including mechanism for both instantaneous start and instantaneous stop by remote control
2. Backspace by remote control for repeating missed words
3. Time scale and interpretation system for quick determination of letter length, corrections, and special information
4. Electronic system comprising the amplifier, listening devices, and playback
5. Special warning and safeguards to assure trouble-free transcription.

Fig. 1—Cutaway section of the capacitor type, fast-starting drive motor gearbox exposes the reduction gears which reduce the speed to 33-1/3 rpm for direct connection to the recording turntable. Beside the motor are shown a gearbox cover and bearing assembly, a 33-1/3 rpm gear, a motor armature and worm, a flexible coupling, and a turntable spindle and worm

Since the two machines comprise a system, it is difficult to separate them, and often a feature is provided on the recorder which has no function except for the result it produces on the transcriber. This resulted in a series of problems each of which often could be solved separately and each solution, combined with other solutions, were co-ordinated into the final equipment.

Mechanical Drive: This problem involved designing a system to give maximum fidelity in a minimum space with simple construction, low maintenance and ease of operation.

SoundScriber developed a small capacitor motor back-geared with a worm drive to 33 1/3 rpm with the worm drive facilities cast in the lightweight motor casting, *Fig.* 1. This small motor is designed for a fast start of the record equal in effectiveness to that of a clutch, and thereby eliminates the need for the usual continuously running motors up to the clutching point. With built-in gearing on the motor, sealed-in lubrication of bearings and gears are provided. The drive is mounted as a single unit obviating belts or other speed-reduction devices, *Fig.* 2. A flexible membrane type of coupling connects the turntable spindle with the slow-speed shaft, and this, together with soft spring supports for the motor,

eliminate vibration from reaching the recording, *Fig.* 3.

This gives the speed-stability characteristic to the turntable which, incidentally, is the heaviest single casting in the machine and the only zinc diecasting. A phenolic top shell is supported on a rubber ring cemented to the heavy casting below to give additional vibration filtering. This two-section structure also provides a hard, smooth surface to support the record and resist the downward pressure of the embossing stylus. Nevertheless the resilient support under this hard surface dampens spurious vertical movements of the recording stylus, *Fig.* 4.

Conventional recorder feed mechanisms have utilized feed screws usually cut with a buttress thread and engaged by a half nut. Such constructions are difficult to enclose and the feed screw requires occasional adjustment for end play. The system used in this recorder employs two worm gears, one of which is engaged with the center spindle and the other at the axis of a radially-swinging recording arm, *Fig.* 5. This structure is completely sealed in grease where it is free from dust and dirt and never needs adjustment. To get "tangential' line-up of the stylus with the groove, the head of the recording arm (and the playback arm as well) is offset in the conventional phonographic manner, *Fig.* 4.

This feed and drive system permits complete sealing of all parts and is an interesting series of speed reductions, *Fig.* 2. The first reduction is from 1750 to 33 1/3 rpm on the slow-speed turntable shaft. The second reduction is also a worm and gear reducing

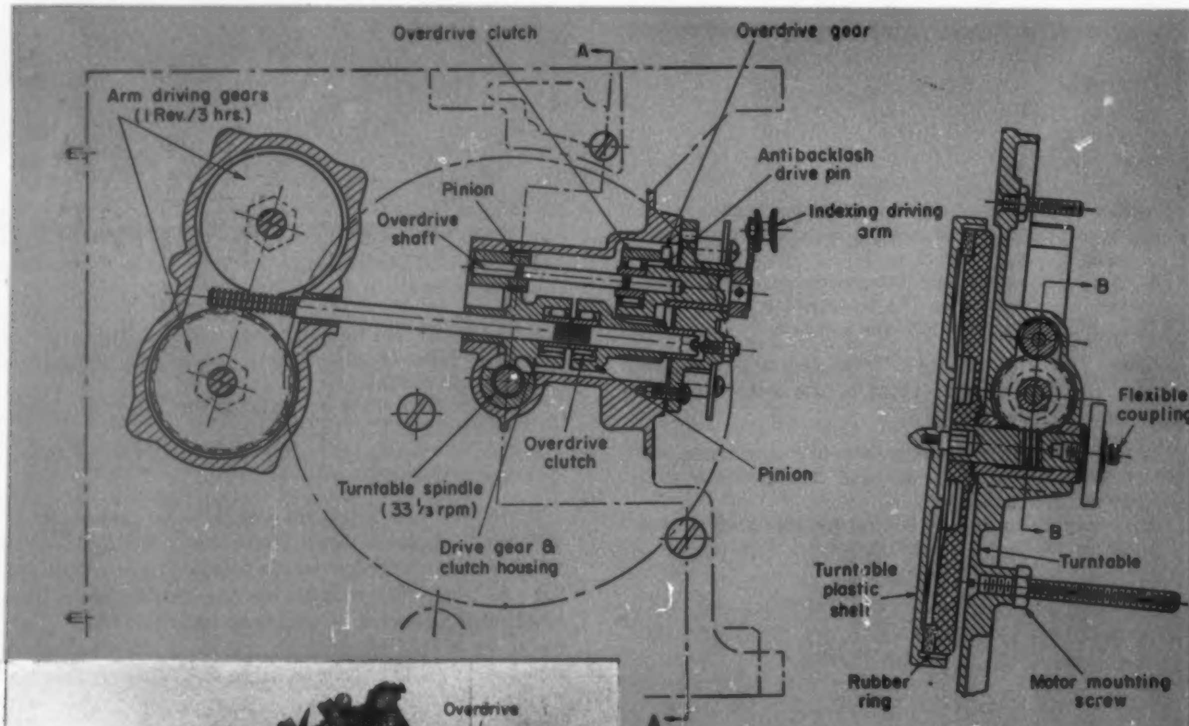


Fig. 2—Above—Drive and overdrive mechanism on the motorboard assembly shows turntable spindle and arm drives as well as the overdrive

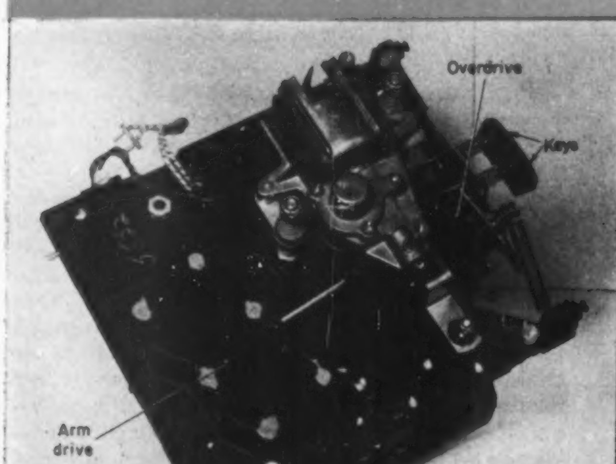


Fig. 3—Left—Bottom view of recorder chassis. A single motorboard casting includes a gearbox to drive the recording and playback arms and an overdrive gearbox to momentarily increase the arm feed for signal indications. The overdrive is controlled by keys

the speed of the intermediate shaft between the turntable spindle and the recording arm axis to 2/3 rpm. The third reduction at the axis of the recording arm reduces the rotational speed of the recording arm as it feeds to about one revolution every three hours. Obviously, with these slow-speed elements, all sealed in grease, wear is not a factor. A trouble-free, wear-free, mechanical mechanism is produced by such a system.

Transcriber Playback Is Free-Floating

In the transcriber, the mechanical drive system is similar excepting that there is no feed system required. A free-floating playback arm, as used in high-quality phonographs, is employed. Refinements — making this arm of extremely lightweight design to have low inertia, with lateral counterbalance to permit off-level operation (as much as 45 degrees), and with a novel spring arrangement to compensate for frictional forces between stylus and disk—free the stylus from unnecessary work in propelling the arm, *Fig.* 6. This gives high fidelity reproduction characteristic of a free-floating playback, allows for simple and accurate backspacing because side play in the mounting of the head is eliminated, and provides ability to track across indexing spirals to be mentioned later.

The transcriber must have an instantaneous stop as well as an instantaneous start. Otherwise, there will be loss of words in transcribing. On the recorder, a short coast is not at all undesirable and any harmful effects are eliminated by having the sound cut off during the short coasting period of the motor. Start-stop on the transcriber, however, is accomplished simply by utilizing the same turnable and center spindle drive system as on the recorder but by supplying the top surface of the heavy turntable with a felt ring on which is supported the same light phenolic turntable shell as used on the recorder. However, in this case the phenolic turntable shell is engaged by a brake shoe which holds it from turning when the brake is magnetically applied by actuation of the typist's switch. When released for starting, the shell gains full speed immediately. The extremely light weight of this phenolic shell gives indefinite life to the felt surface and as close to an instantaneous start-stop as possible.

Controls Are Simplified

PLAYBACK: The second major problem required provisions for a simplified listen-back system, eliminating irritations that cause loss of thought due to distracting operator manipulation.

The dictating machine must have a start-stop, whether it be on a hand microphone, a foot control, or some form of hand control, because the operator requires time intervals to think and the record must be stopped during these periods. A study of the listen-back process revealed that there are really two different kinds of listening. The first is the type of listening back which the dictator does in order to recall his last words after interruption (quick review).

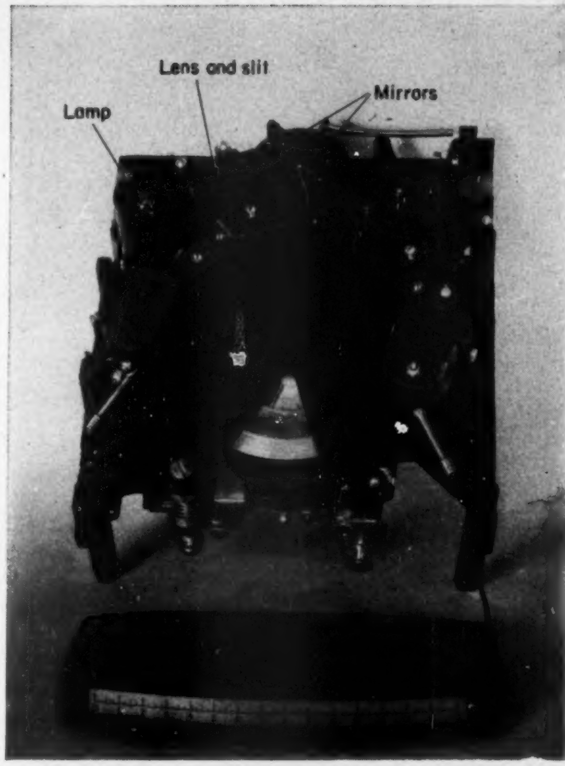


Fig. 4—Recorder time scale system. Cutaway section of the recorder turntable shell exposes the heavy cast-zinc turntable and the sponge rubber ring on which the shell rests. Position of the arms is indicated on the translucent scale by vertical beams of light originating from a lamp, lens and slit assembly and reflecting from mirrors to focus on the scale screen. Mirrors are driven by the arms, causing the light beam to register position on the scale

The second is when he wishes to listen back to earlier paragraphs for purposes of reviewing the completeness of his dictation or editing.

It has been conventional practice to have the recording stylus and playback stylus mounted in the same moving assembly and separated by a distance equal to a few grooves. For playback, a control on the recorder lifted the recording stylus, lowered the playback stylus, and simultaneously selected the proper circuits for playback or recording. In lifting the recording stylus from the record, the groove was interrupted and these breaks in the groove have always been responsible for a fair percentage of the trouble experienced in transcribing. This system further gave no adjustability of the amount of this quick review to the listener's habits.

A novel system was devised to overcome these difficulties based on having the recording and playback heads completely separated and independent. By having both recording and playback styli on the record at the same time with an adjustable amount of trailing of the playback stylus, listening to the last words requires only the selection of necessary circuits controlled remotely at the microphone. Furthermore, the recording groove is not interrupted by this quick review process since the recording stylus

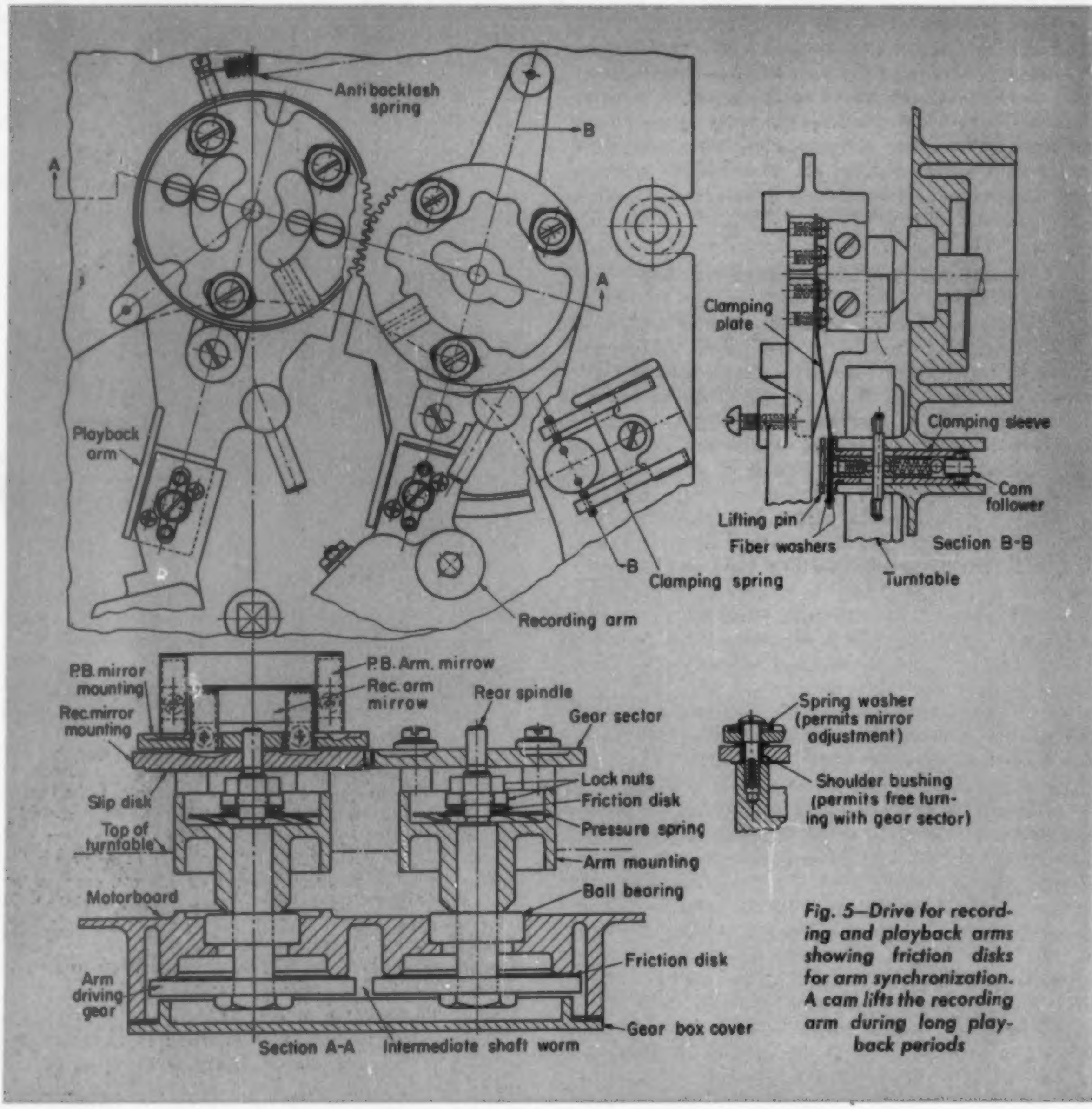


Fig. 5—Drive for recording and playback arms showing friction disks for arm synchronization. A cam lifts the recording arm during long playback periods

is not lifted. Blank unrecorded grooves are created while listening instead of the equal amount of ungrooved blank area resulting if the stylus were lifted. The electrical actuation of the talk-listen circuits from the microphone itself frees the dictator from the need of touching the machine at all in order to recall the last words he had spoken. This actuation requires an additional switch on the microphone called the quick review switch, shown in *Fig.* 7. To play back last words (quick review) the dictator pushes in the button and pulls it down, thus actuating the "quick review" switch. On returning to dictate, he presses in the normal manner. Thought is an extremely transient thing and the ability to immediately resume dictating after listening back without distracting movements has proved extremely valuable.

This same general system lends itself very well to the second type of listen-back in that the separate recording head can be locked in position when it is lifted from the disk for a long listening period. When the dictator is through listening, he turns the talk-listen knob to "talk" and the recording stylus returns to the record in the correct place without the usual precautions to avoid the hazard of recording over previously recorded grooves. This lift and lock mechanism is shown in *Fig.* 8.

The means of setting up trailing and synchronization automatically of the styli will be discussed later.

INDEXING: Another interesting problem was to design magnified time scales and an indexing system for easy operation and foolproof transmission to the secretarial desk.

Conventional systems have employed cards or slips which generally provided the printed time scale as well as a system of indexing by marking on the slips.

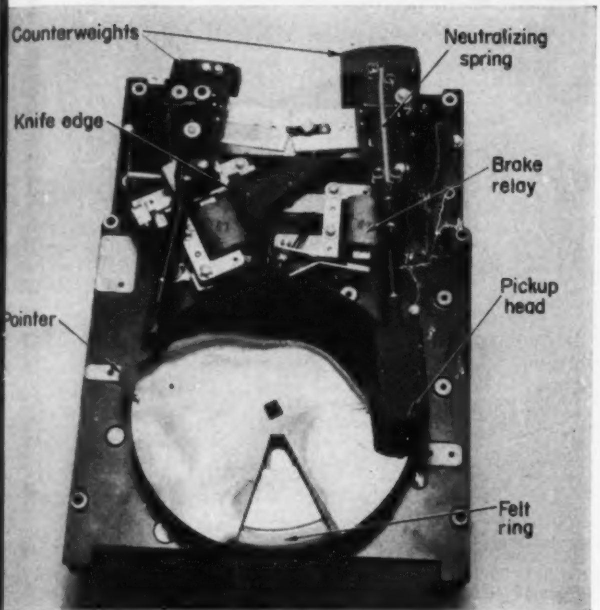


Fig. 6—Pointer arm for transcriber is geared to the playback arm to provide accurate indication of stylus position. Brake permits instant stop and start of turntable shell

On plastic records, the playing time is consolidated into a small space and this introduces special problems of precision. The combination of slow speed of rotation and fine grooving results in a relatively long recording compressed into a small spacè, thus calling for considerable magnification in order to provide indexing accuracy.

Mirrors Indicate Synchronization

The method of producing the visible time scale utilizes optics. The principle of rotating a mirror at the axis of the arm gives the magnification needed by projection of a light beam on the time scale. The time scale, therefore, shows the position of each of the independent styli on the disk. This provides an improtant adjunct to the listen-back system because, when the dictator has listened back a long way, the two styli are out of synchronism thus destroying the quick-review feature. By merely moving the playback arm, however, until the light beams superpose, synchronism is again achieved, *Fig*. 4.

The light beam indexing system of magnifying and accurate automatic indexing emphasize necessity for precision since gear errors, spindle runout, errors in position of styli, etc., become critical. Place finding, indexing and backspacing problems are accentuated in this compact recording. Magnification and exacting precision in all the mechanical and optical parts provide the solutions.

The disk record is unique in that it provides a large open area on which indications may be made automatically to show both length and correction signals. Many experiments were made with different kinds of pushbutton actuated markings on the disk surface,

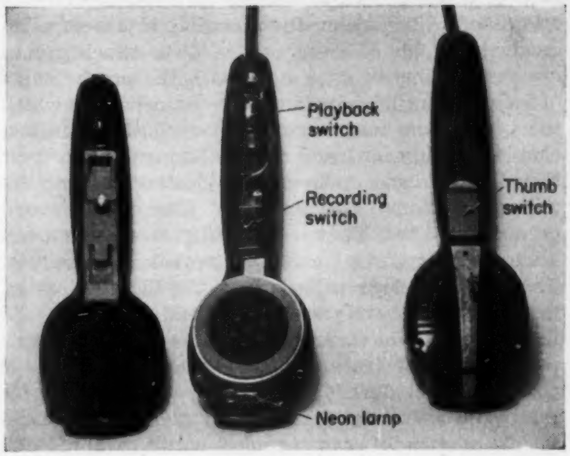


Fig. 7—Above—Microphone and switches are enclosed in a molded plastic case. The moving coil type microphone also serves as a reproducer for playback. Neon lamp glows when controls are set for recording

Fig. 8—Below—Mechanical chassis of the recorder is a complete unit and separate from the amplifier chassis. Recording arm lift and clamp mechanism prevents feeding while portions of the recording are being played back

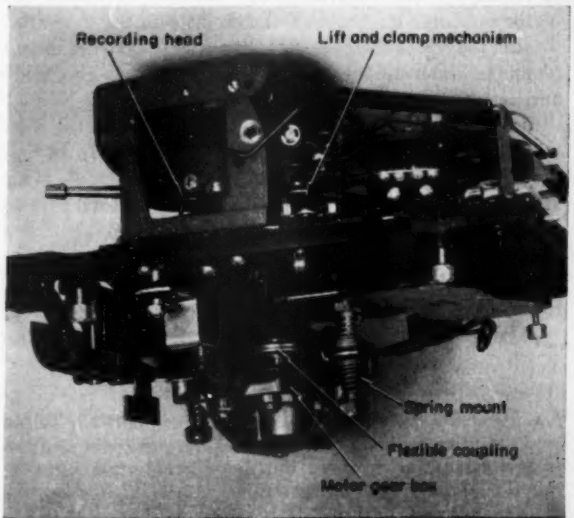
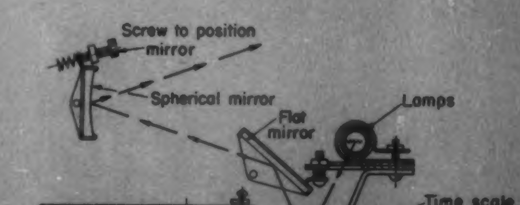


Fig. 9—Optical system of transcriber. Time scale and lamps are mounted on hinged front cover and are lifted out of the way with lid for loading disks



none of which proved successful until the method was originated of advancing the recording stylus so as to create small blank areas on the disk. Such spaces are very visible in good light with the proper angle of viewing, but a means had to be conceived by which the typist, who may have a certain hostility to mechanical equipment could quickly and easily interpret them. The conception of a reflecting magnifying mirror to show the specially illuminated sound grooves and the blank contrasting areas against a scale on the typist's transcriber provided the answer. Thus, the spacings indicated by the dictator are on the disk itself—wide spaces for end of letters and narrow spaces for corrections—and provide a mistake-proof means of transmission of these indications to the secretarial desk. The brilliant projection of the images in the magnifying mirror gives the typist a visible system of unquestioned accuracy for locating corrections and ends of letters. The arrangement of this system is shown in *Figs.* 9 and 10.

Provisions for Judging Length

The time scale on the transcribing machine is a mechanical scale reflected in the mirror along with the disk, permitting the typist to judge the length of letter by the length of the distance between the blank spacings on the disk itself. A pointer is geared to the playback arm by feather-touch gearing freed from backlash so the pointer travels across the scale and is reflected with it in the mirror.

Method of producing these blank spaces on the disk also provided some interesting problems. It is important that the playback stylus on the transcriber move from one blank area to the next smoothly and quickly. As mentioned before, a gear-fed playback would not do this. The free-floating playback, operating in a connecting groove between the groove sections, gave the desired result on the transcriber.

It will be recalled that there is a transmission shaft connecting the center spindle worm gear and the rear spindle. Into this connection a gear transmission is introduced which permits a 4 to 1 overdrive, *Fig.* 2, put in at will by pressing the proper key, thus giving a momentary coarse feed to the stylus arms and a highly visible spacing. The length of time that this coarse feed is applied determines the width of the space, so that for corrections it is made small, and for ends of letters it is made larger. For special indications, calling for typist attention before any other record material is played, such as for additional carbon copies, etc., both keys may be depressed at once making an extra large space easily identifiable from the other spaces.

This transmission is shown in *Fig.* 11 and consists of an overdrive and a pacing mechanism. Depressing the end-of-letter key stores up power in a spring and the overdrive clutches of the roller type engage to overdrive until the key returns to its normal position. The stroke is less when the correction key is depressed giving a smaller spacing. When both are depressed the strokes are additive giving a spacing equal to the sum of the spacings.

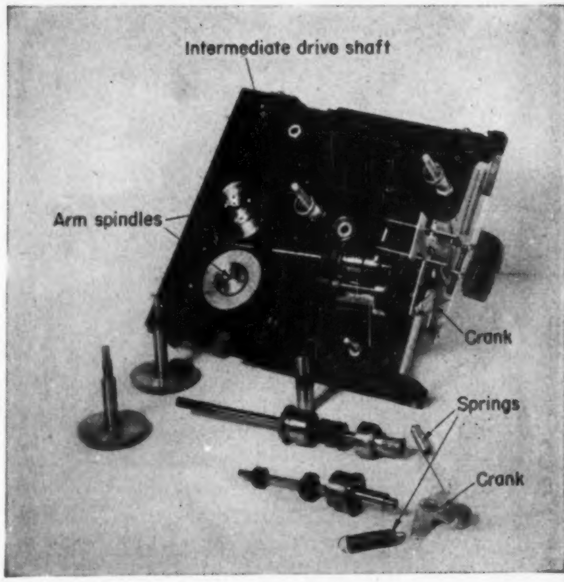


Fig. 11—Cutaway section of the overdrive gearbox and open rear gearbox exposes the parts which drive the arms of the recorder. Similar rotating parts are opened up in the foreground to illustrate the overdrive clutches and gears in the overdrive mechanism. Operation of the index keys turn the crank a gaged amount. Springs then momentarily take over the drive of the feed system at a 4 to 1 ratio above normal. The motor only paces the action

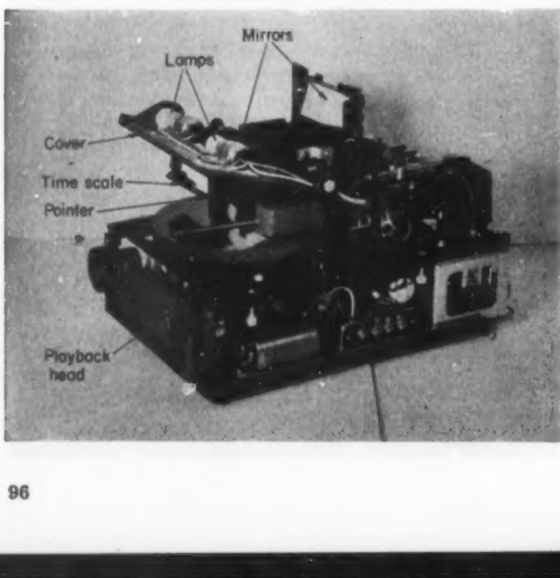


Fig. 10—Transcriber with the loading cover partially raised and the arms partially retracted. When the cover is down in the operating position, the index scale rests close to the disk and is lined up with the pointer, indicating the exact position of the playback head

ELECTRONIC AMPLIFIER: To design an electronic amplifier system for these machines required special emphasis on dissipation of heat and easy accessibility, capable of being serviced by mechanical technicians.

The electronic amplifier is built completely in a lower chassis where it can be harnessed together with the controls. A compartmented section in the back separates a plug-in card amplifier which may be replaced by unskilled technicians with no difficulty. The compartmenting of the power tubes confines the

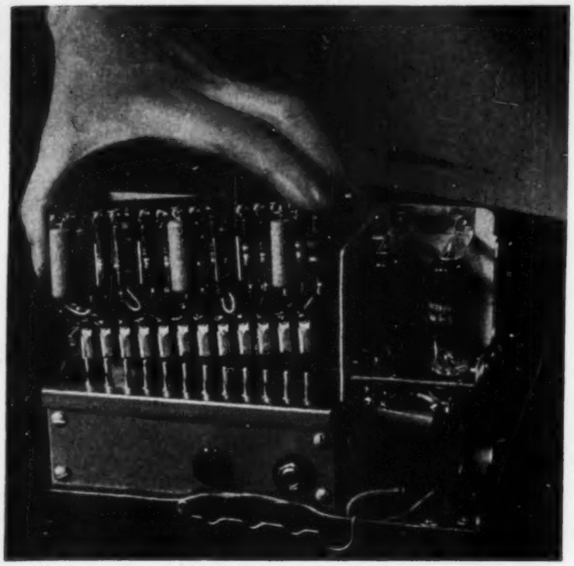


Fig. 12—Rear view of recorder showing plug-in card amplifier being replaced. Sectionalized construction of chassis isolates high operating temperature components from other elements operating at low temperature. Open grilles provide ventilation

main source of heat in one corner where the compartments protect such elements as selenium rectifiers and electrolytic condensers from heat, *Fig.* 12. This eliminates the need for continuously running motors to give forced ventilation, thereby preventing a great deal of dust and dirt from getting into the interior of the equipment.

In order to further extend the mechanical principle of eliminating all mechanical motion or wear during the standby periods by the use of the motor start rather than a clutch system, the electronic system was so arranged that the start-stop relay has extra contacts so the tubes draw only about 20 per cent of normal plate current during the standby period, thus assuring long life of the system.

An additional feature of interest is the muting switch on the transcriber which short circuits the output to the listening device whenever the brake is on, eliminating any extraneous hum or other noises from the listening device.

Loading Features: Designing a system for putting on and taking off records involved the problem of providing safeguards to facilitate use and prevent errors.

In the loading, or reloading of a record on a dictation machine, certain steps are always necessary. First, the recording stylus must be lifted from the record before it may be moved. Next, the recording stylus must be moved from its finishing location to the starting position of a new record. Thirdly, the record is changed, and fourthly, it is necessary that the recording stylus be placed against the record at the start of the new record. Loading a dictation machine, including handling of index slips, has been a cumbersome process and one which a user has had to learn thoroughly to avoid misoperation of the machine, resulting in a poor or spoiled recording.

The recorder is designed to make this procedure as simple and automatic as possible. This is accomplished through the use of a hinged top cover, which, when it is down, serves as a protection for the record, in normal position. It also contains the time scale. By simply lifting this cover the recording stylus is automatically raised, the recording arm and playback arm are automatically returned to their respective starting positions, and the record is accessible for quick removal and replacement, *Figs.* 13 and 14. After the record is placed on the turntable, the act of closing the cover insures that all the proper controls are set for recording. No manipulation of controls is necessary in changing records. Both time and effort are thus kept at a minimum. What is more important, perhaps, than time saved, is that loading becomes so automatic that the user avoids errors which might either obliterate or destroy a recording, or to start a new recording with some misplaced control.

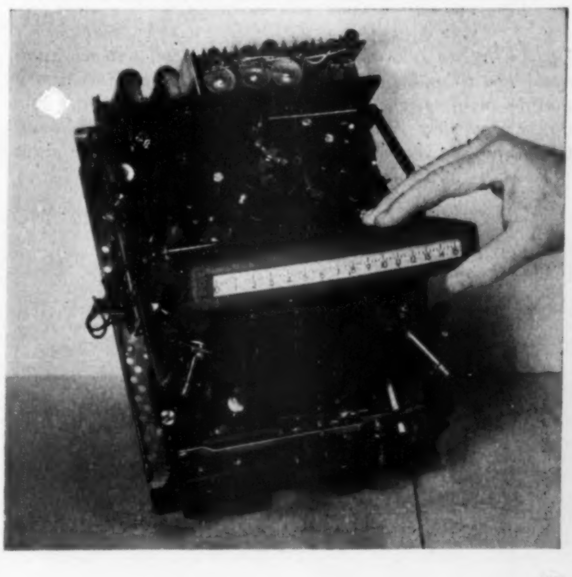
Design Is Compact

The machine is designed to use minimum desk space—no more than the base of the machine. Being a "top loader", it does not require any additional space or clearance for the loading operation or use.

While 15-minute disks provide for the bulk of dictation, there is need frequently for shorter-length records. A 3⅞-inch disk gives playing time of 7½ minutes on each side and a 3-inch size plays 4 minutes on each side. Both of these smaller sizes will go into ordinary envelopes for mailing.

An interesting fact is that the playing time and sizes of these disks are so balanced that 7½ minutes can be recorded on the 3⅞-inch disk and re-recorded or copied on the same machine on the outer area of a 5⅞-inch disk. The ability of the machine to use this "disk on disk" feature provides an important

Fig. 13—Lifting the cover retracts the arms to the beginning position and exposes the turntable for changing disks



facility in using these smaller disks.

SPECIAL SAFEGUARDS: Warnings and safeguards were required to assure freedom from trouble at the transcribing end.

A warning signal at the end of the record is fundamental. It was decided that this warning should continue with each rotation of the turntable until the dictator has done something about it. The solution was in having a projecting portion of the turntable shell engage a member attached to the recording arm in such manner that a bell is struck at each rotation of the turntable after the $14\frac{1}{2}$-minute point of a record is reached.

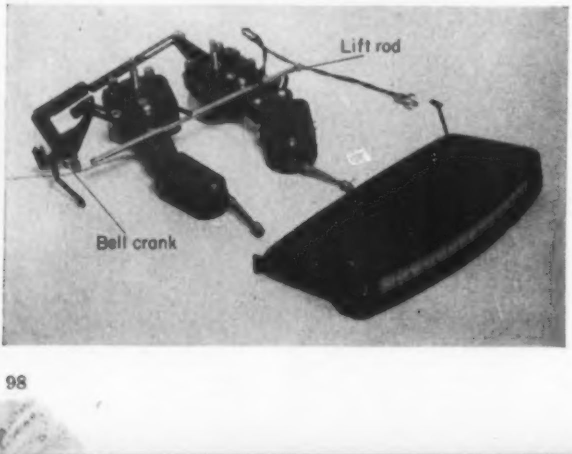
Prevention of scratching of records during loading or take-off has always been important. The method of automatically lifting and retracting the heads by the hinge cover at the front completely eliminates this possibility in ordinary use. Also, the extremely light pressure of the transcribing stylus makes possible skidding it across the the sound grooves without harming the record. This provides an important ability to quickly scan the record to find material without harming the recording.

A neon light on the microphone is arranged to operate in series with a switch section on the function selector at the left-hand side of the machine and the listen-talk selector in such manner that the function selector must point to "dictate" and the listen-talk selector must be at "talk" for the neon light to be on. In other words, if the neon light is not on, the machine is not ready for recording.

Flashing Lamp Indicates Recording

At the front of the machine is a lamp of the flashlight type connected across the recording head. This head is low impedance so that the voltages and currents developed are sufficient to flash the light, its intensity varying with the intensity of the signal. Accordingly, it serves as a good guide to the proper volume for dictation, having considerably more gradation than that of a neon light. It also serves as an indication that dictation is being properly recorded because it does not flash in any position other than normal recording.

Fig. 14—When reloading, the cover is raised and the arms are lifted from the disk by the rod and retracted to the outside of the turntable against adjustable stops to begin a new recording. The bell crank is connected to the cover. Links and associated overtravel springs retract the arms to their outside stops



Another interesting feature is the method of introducing the power cord to the machine. The cord has two wires going into a 6-prong socket which engages a 6-prong male socket on the recorder and on the transcriber. The purpose of the six prongs is to provide connections for direct-current operation from an inverter. Turning on the one switch on the transcriber powers the inverter, thus eliminating the need for two switches. On the recorder, circuits are arranged so that the inverter only operates when the motor runs. This comes about because of the ac-dc design of the amplifier and the need for alternating-current for the motor. The panel for these plug connections is in the bottom of the machine and is recessed so as to allow room for the plugs. The wires are brought through channels pressed out of the base plate, having a catch for securing the cords in place.

Spring Eliminates "Double Talk"

Embossed groove recording has presented special problems since no material is removed when the groove is formed and piles appear on both sides of the groove walls. This effectively creates a groove between grooves and "double talk" results if the playback stylus lands in this area and gets vibratory impulses from each pile. The free-floating playback arm eliminates any "synchronism" between the playback stylus and the groove on the transcriber and therefore gives far less tendency to remain in this "groove between grooves" than if a gear-fed playback were used. A forward urge by means of the compensating spring on the playback arm of the transcriber causes the stylus to drift rapidly from any one groove section to the next even though they may not be joined by the characteristic grooving of the deliberately produced spaces in indexing. This further gives a tendency to overcome a double-talk situation.

Grooved Spacings Aid Transcription

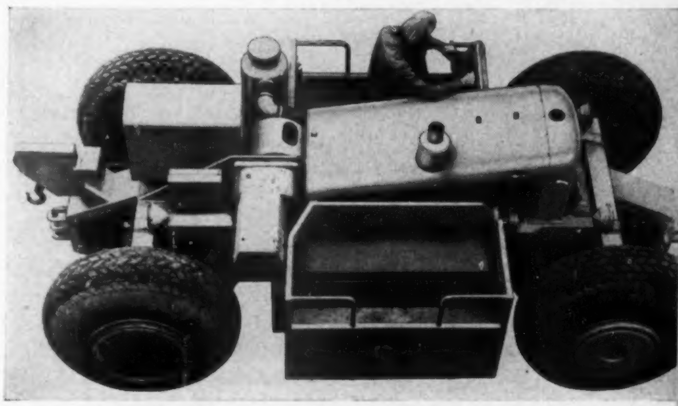
The continuous groove formed during the quick review operation is a decided asset to transcribing in that it eliminates a blank area. Blank areas, however, are produced when the groove is broken in the long listen-back period. This means that the playback stylus, in drifting across, can engage the first groove of the next section at any point of its periphery and may therefore lose some of the first words if they are spoken immediately by the dictator. To prevent this possibility, a safety or protective groove is automatically created on the recorder every time the dictator swings from "listen" to "talk". This is done by the capacitative holding principle of a relay which permits the motor to continue running until this capacitor is discharged. The time rate is set for a protective groove of about one revolution.

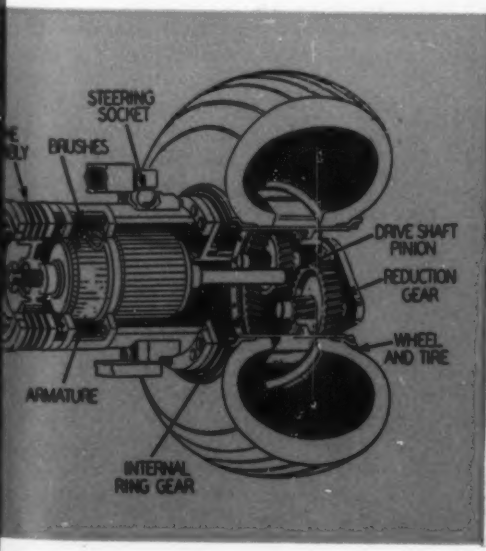
Solutions of the design problems, discussed, co-ordinating them into a pair of attractive and efficient machines, have eliminated many of the annoyances that previously had plagued the industry.

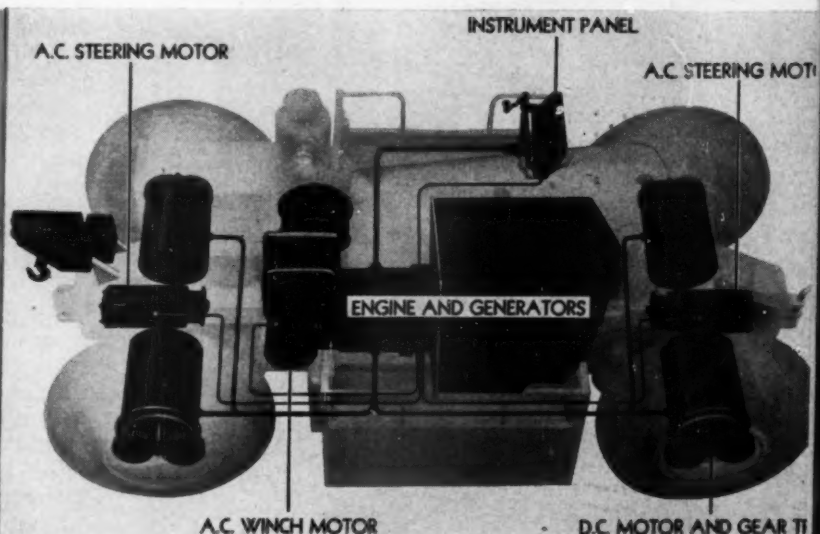
SCANNING the Field For Ideas

Individual electric drives on each wheel of the heavy-duty tractor for earth moving equipment, below, obviate clutch, differential and conventional transmission. Designed by R. G. LeTourneau Inc., this tractor is powered by a 6-cylinder 2-cycle diesel driving a direct-current generator for supplying power to each of the four drive wheels as shown in the diagrammatic phantom view below. The power is applied indirectly through a potentiometer type rheostat for precision control. Also driven by the engine is an alternating-current generator, in line with the d-c generator, for powering two steering motors for front and back wheels and a motor for operating a winch on the rear of the unit.

Each drive wheel is suspended on ball and socket type bearings with drive motor, reduction gear train, drive gear, and wheel built as a single unit, below left. The d-c motors provide high starting torque, wide range of speed control and automatic acceleration under momentarily increasing load conditions. Their characteristic regenerative braking is used to advantage in retarding the load on grades, eliminating the necessity for a separate braking system during operation. A parking brake, however, is built onto the motor to lock the armature when no current is applied to the motor. This brake is a multiple-disk, magnetic-release, spring - engaged unit mounted on the rear of the motor frame. As long as current is flowing in the system, the brake is disen-







gaged. When current is cut off the springs engage the brake, holding the load in positon until power is applied again.

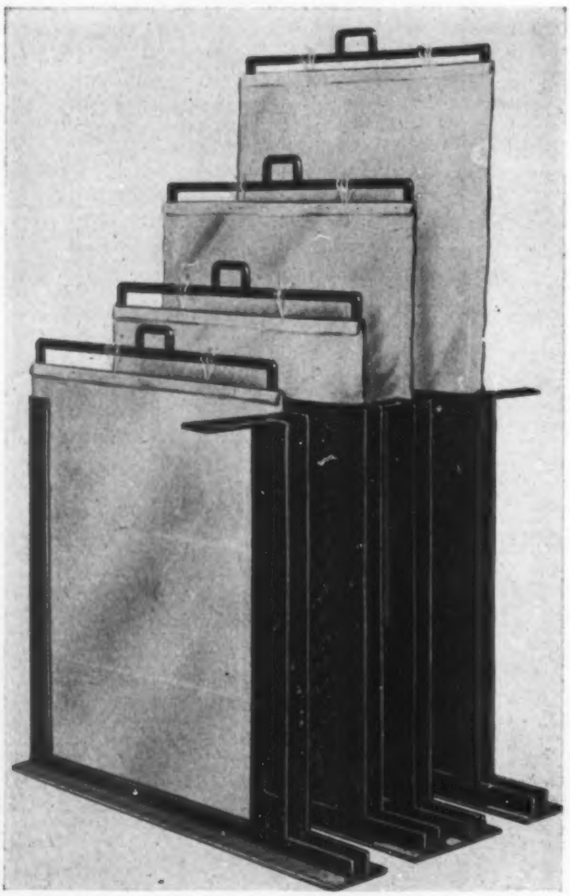
Induction motors are employed for steering and for the winch because of their starting, stopping and reversing characteristics. To obtain high starting torques, the motors have built-in rotor resistance in the form of a stamped, stainless-steel resistance bell. Copper rotor bars extend through the rotor laminations, are flared outward and upward, and are welded to the bell.

On the opposite end of the rotor these bars are welded to a low-resistance iron ring. The combination provides a variable resistance by taking advantage of the unusually high temperature coefficient of resistance of stainless steel. When the motor is started under heavy load, high induced rotor current causes the stainless steel bell to heat rapidly, resulting in a high starting torque. The effect is the same as the insertion of a starting resistance in a wound-rotor motor. In starting heavy loads, the bell actually may get red hot without damage to either rotor or stator because the heat is generated beyond the windings. As speed increases, air cooling reduces the temperature of the bell, decreasing resistance and increasing efficiency.

Potentiometer-controlled generator-field excitation is provided from the a-c generator through step-down transformers and selenium rectifiers. These potentiometers not only control the speed of the wheel motors but also reverse their direction without use of circuit breakers. By this method it is possible to control the high amperages in the feeder lines by controlling low amperages in the field excitation, keeping high currents completely apart from the controls.

In the a-c system, a constant-voltage transformer is in the generator excitation circuit to maintain constant voltage at the generator regardless of the varying loads experienced during the operation cycle.

Diaphragm plating using canvas duck screens for the segregation of anode and cathode compartments—produces smoother plated surfaces, allows for higher current densities and eliminates polarization.

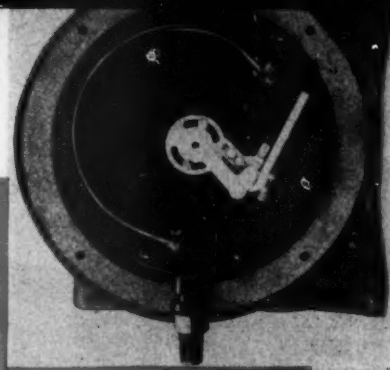


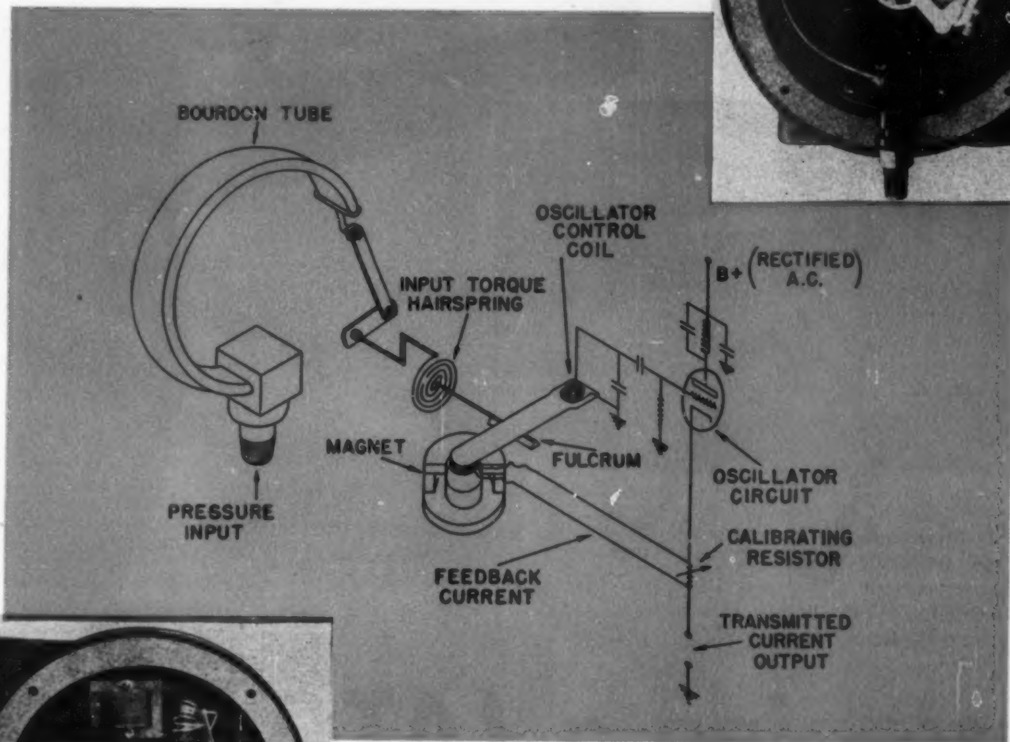
The anode solution is pumped off, filtered and passed to the cathode compartment free of insolubles which form at the anode and ordinarily deposit on the work pieces. The cost of equipping for this method of plating, however, has been its chief drawback. Where size of work permits, standard tanks may be equipped with the unit illustrated at left. Developed by Automotive Rubber Co. the unit employs fabric sleeves sliding on rubber insulated metal frames which in turn slide into insulated channels with the cathode side or work lane protected by insulated woven wire guards.

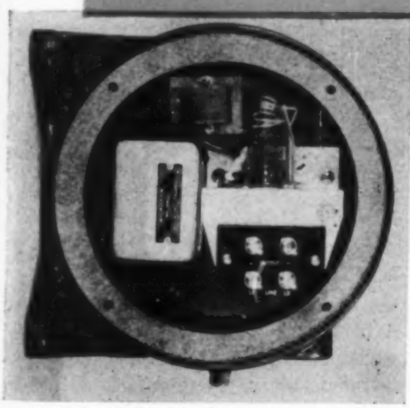
Transmittal of pressure data from various remote locations is simplified with the unit shown at top of next page. This transmitter is located at the source of pressure to be measured and is connected to an indicating or recording device by a simple two-wire circuit, obviating extended high-pressure lines for this purpose.

Developed by Manning, Maxwell and Moore Inc., the Microsen transmitter employs a tuned oscillator circuit, the tuning of which is affected by a pressure-sensitive element. This element produces a mechanical balance between the applied pressure and the output signal. A pressure change modifies the deflection of a Bourdon tube and linkage converts the travel of the tube tip to rotary motion of a shaft which is connected through a hair spring to a beam. Beam movement detunes the oscillator circuit, causing a change in plate current through the load resistance. A portion of this current is fed back to a coil on one end of the beam and in the field of a permanent magnet, stabilizing the beam at the signal output level.

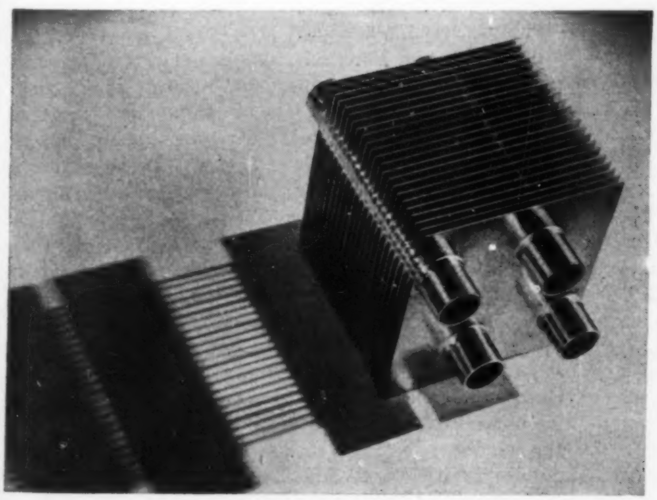
Ambient temperature variations are eliminated in the system. Temperature effects on the Bourdon tube





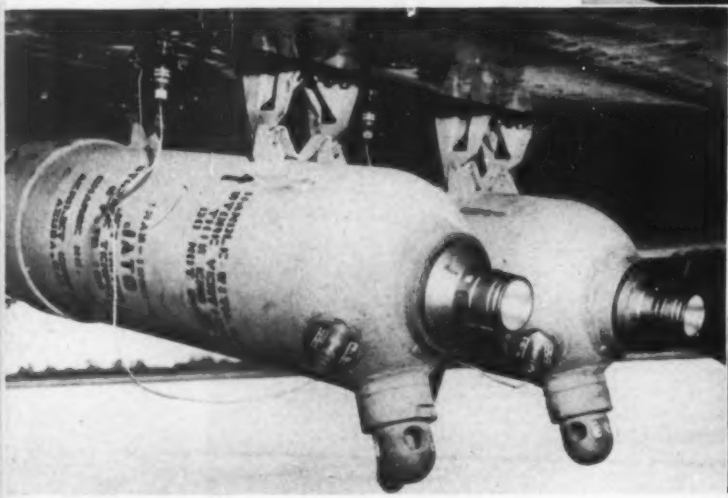


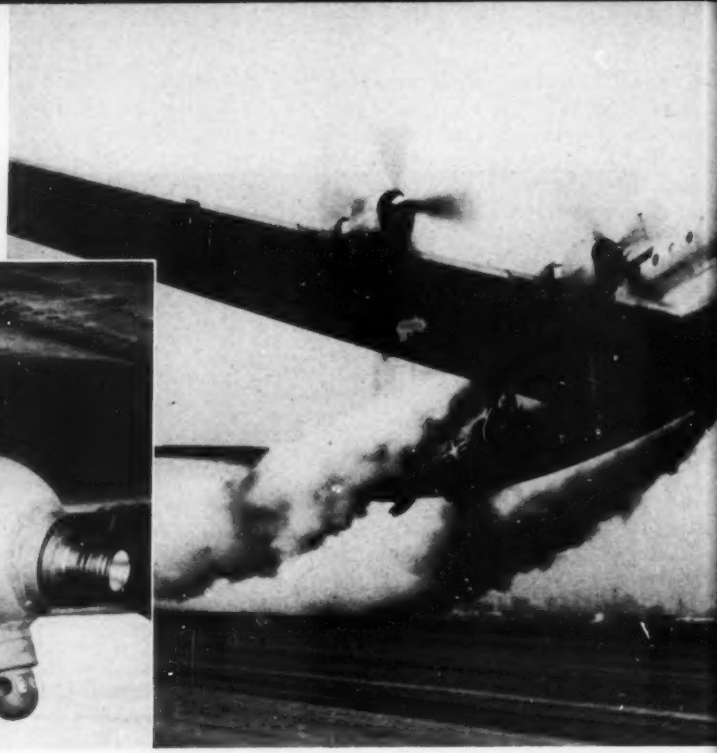
are opposed and nearly equalled by the change in the modulus of elasticity of the torsion spring. Final compensation is achieved by temperature responsive resistors in the feedback network. Seminull balance principle of operation automatically compensates for the variations caused by electronic tube deterioration, variations in line voltage, or changes in line resistance. The transmitter, top and left, is compact and is designed to withstand accelerations of 30*g*.



Mechanical bonding of fins to tubing in heat exchangers, section shown at right, obviates welding or soldering operations in assembly. Flat plate-type fins are punched so as to form a bonding collar at each hole, providing a broad surface of contact with the tube instead of a sharp edge. In assembly, after fins are in position on the tubes, a tool-steel bullet is forced through the tubes by compressed gas, expanding the tubing uniformly against the continuous surface formed by the collars of the fins. Carpenter steel tubing is employed in these corrosionproof units built by the Trane Co. The mechanical bond withstands repeated cycles of expansion and contraction, making it adaptable to the heating and cooling cycles in air conditioning equipment.

Fig. 1—Below—Jet assist take-off rocket motors mounted and wired for firing





Designing a JATO Engine

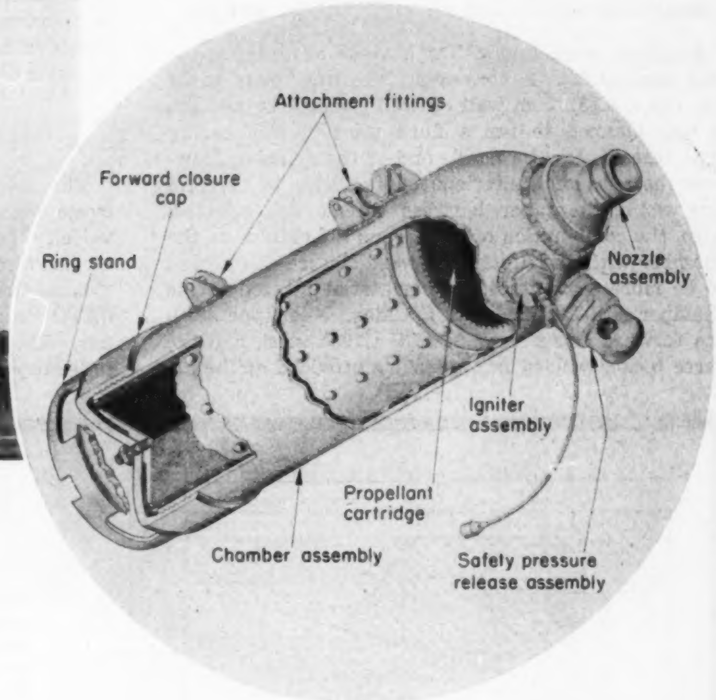
PRIMARY factors guiding the design of a jet-assist rocket engine are reliability and low cost. Reliability, which implies close manufacturing control, is of principal importance because a pilot may be committed to a take-off when he fires the rocket engine, without which the take-off could not be accomplished. Low cost is of particular importance because in tactical operations the rocket is jettisoned after it has performed its function. Successful incorporation of these two factors into the design of a rocket engine can be illustrated by a study of the Navy's MK 2 MOD 3 (14AS-1000) JATO, *Fig.* 1. Developed by the Aerojet Engineering Corp., this unit incorporates results of basic research conducted at the Guggenheim Aeronautical Laboratory of the California Institute of Technology (Galcit). Although the design was completed in 1945, the rocket is still one of the most used in jet assist take-off of aircraft, and has the distinction of being the first rocket engine awarded a type certificate by the Civil Aeronautics Administration.

A cutaway view of the JATO is shown in *Fig.* 2. In operation, the propellant cartridge burns back along its axis like a cigarette, the resulting gases exhausting through the nozzle and producing a thrust of 1000 lb for a duration of 14 seconds. In construction, the nozzle, igniter and safety pressure release assemblies are threaded into bosses which are welded into the aft end of the chamber assembly; attachment lugs are welded to the cylindrical section of the chamber. The forward end of the chamber assembly receives the propellant cartridge; the forward closure cap is assembled to the chamber by threading. This article outlines the design development of these basic rocket engine components treating in some detail the more critical elements.

Propellant Cartridge: The power source of the rocket engine is a cylinder of propellant coated with an inhibitor which confines the burning of the propellant to the open end. The propellant is a mixture of powdered potassium perchlorate (oxidizer) and an asphalt-oil mixture (fuel). The two are mixed at a temperature of 290 F, when the fuel is in the liquid state. When mixing is complete, the propellant is cast in molds and cooled to solidification.

The inhibitor coating, an asphalt-oil mixture, is then applied, and the completed charge is wrapped in tape. An aluminum disk is taped to the closed end of the charge; the disk is threaded for a stud which secures the charge to the forward closure cap. The charge is encased in a 16-gage terne plate sheath which is dimpled to provide contact with the inside of the chamber and yet leave a passage around the charge for the rapid equalization of pressure at igni-

Fig. 2—Cutaway view of the JATO rocket motor. In operation, the propellant cartridge burns back along its axis like a cigarette, the resulting gases exhausting through the nozzle and producing a thrust of 1000 pounds for 14 seconds



By W. L. Rogers
Assistant Chief Engineer
Solid Engine Department
Aerojet Engineering Corp.
Azusa, Calif.

tion. This sheath is secured to the charge with steel strap banding.

Predecessors of this rocket utilized a propellant charge cast directly into the steel chamber assembly, which was coated with an inhibitive material. Because of the difference between the coefficients of thermal expansion of the steel and the propellant, cracks developed in the bond between the propellant and the inhibitor coating at low temperatures. Such cracks exposed additional propellant area to burning and resulted in excessive pressures and consequent malfunction of the rocket. Development of a separate charge not bonded to the chamber wall and a propellant with improved low-temperature properties resulted in extending the lower operating temperature limit of the rocket from +40 F to −10 F.

Chamber Assembly: Experimentation with the solid propellant yields a plot of the equilibrium chamber pressure versus the ratio of burning surface to nozzle throat area, and a plot of burning rate normal to the burning surface versus equilibrium chamber pressure. From these data and from the known relation beween thrust and the product of chamber pressure and nozzle throat area it is possible to determine the length, diameter, and operating pressure of the chamber. However, determination of chamber proof pressure involves additional factors—variation of equilibrium pressure, safety pressure release and strength of the chamber material.

Value of the equilibrium pressure increases with charge temperature during the burning period. For the propellant considered in this article, the variation is on the order of 0.35 per cent per degree F. This "temperature coefficient"—the expected variation in operating pressure from engine to engine—and the basic or nominal operating pressure determined at 60 F are used to determine the maximum expected operating pressure.

The safety pressure release is a simple copper diaphragm closing an orifice in the chamber. The diaphragm is designed to fail within a predetermined pressure range in the event of malfunction of the propellant charge, thus providing additional area for the escape of gases and protection of the chamber against excessive pressures. Magnitude of the pressure range is established from experience, the absolute value of the pressure being established in excess of the maximum normal operating pressure at the upper temperature limit.

Strength of the chamber material decreases as a result of heating during firing and this reduction must be considered.

Proof pressure for the vessel can be established from consideration of the foregoing factors. In this

design the proof pressure applies a hoop stress to the chamber wall close to the minimum yield strength of the material.

For the rocket engine under study, the pressure values are as follows:

Nominal operating pressure (60 F)	2000 psi
Maximum operating pressure (140 F)	3000 psi
Safety pressure release range	3200-3700 psi
Proof pressure	4500 psi

Seamless steel tubing has obvious advantages for this application. A convenient size was found to be 9⅝ OD x 0.281 inch wall oil-well casing. Originally, it was planned to use a flush joint oil-well casing type thread near the nozzle end of the cylinder. However, because of limitations on facilities to perform this work, the taper buttress thread was selected. Also, the combination of a hot spun aft closure on the chamber with an elliptical forward closure cap proved to be more economical. The material utilized in the chamber is a modified API Standard N-80 type steel, the forward closure cap is AISI C1035 steel, and the three bosses welded into openings provided at the aft end of the chamber body are AISI C1015 steel. Attachment lugs which provide three points for suspension are welded to the outside of the chamber.

Nozzle Assembly: A problem common to the design of three components for the JATO engine—nozzle, safety assembly and the igniter—was that of resistance to heat generated by the burning of the propellant charge. Flame temperature of the propellant charge is in excess of 3000 F; however, this temperature must be withstood for only 14 seconds—duration of the rocket run.

Nozzle Throat Major Design Problem

The nozzle assembly is particularly critical because high gas temperature is combined with high-velocity flow. The most critical condition is found at the throat of the nozzle. Erosion first occurs at this section of an all-metal nozzle. Solid propellant JATO nozzles were originally made from solid copper. The design was based upon the mass of copper necessary to absorb the heat transmitted to the nozzle

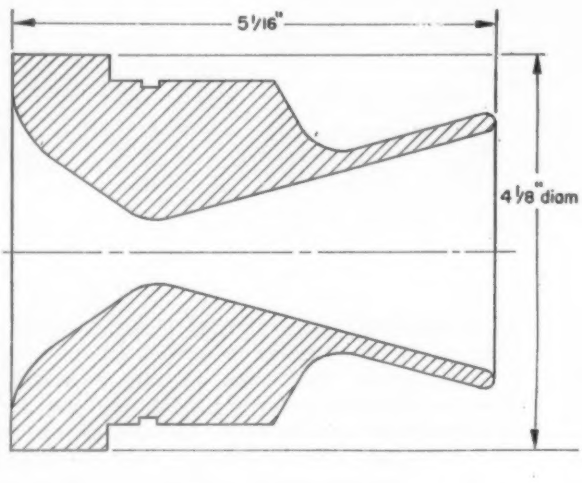
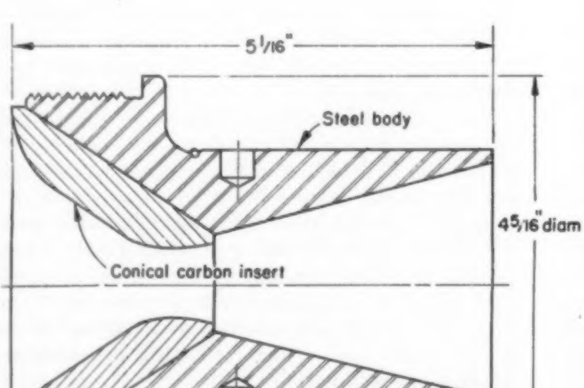
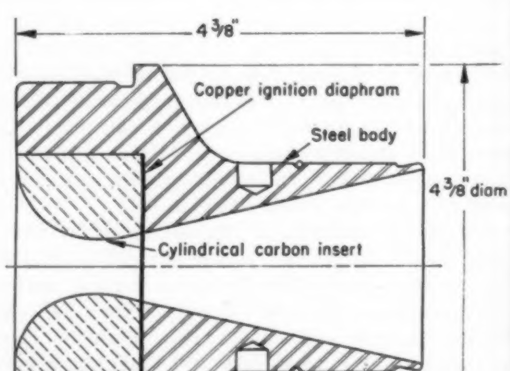


Fig. 3—Left—Solid copper nozzle for a 12-second motor. Size and weight would assume tremendous proportions on 30-second motors

Fig. 4—Left below—Although better than the solid copper nozzle in Fig. 3, this conical carbon-insert nozzle failed at the thin carbon section just downstream of the throat. Breaking of the carbon would proceed forward of the throat causing a drop in chamber pressure and in thrust

Fig. 5—Below—Nozzle design in use today. Defects of the solid copper and conical carbon-insert nozzles have been eliminated by the introduction of a shrink-fitted cylindrical insert and a copper diaphragm which insures ignition reproducibility by maintaining chamber closure until the pressure has reached 500 psi





during the rocket run. The copper nozzle, *Fig.* 3, though heavy enough on a 12-second duration engine, assumed tremendous proportions on engines of 30 seconds duration. Accordingly, a search was made for a material suitable for a lightweight nozzle that would withstand the severe conditions of temperature and gas erosion. Various ceramic materials were tried without success. A tungsten carbide nozzle failed by cracking under thermal shock.

Finally, a conical nozzle insert, machined from extruded graphite bar and held in a steel body, was tried and operated successfully for 30 seconds in a 1000-lb thrust engine. Similar inserts were successfully used in engines up to 2500-lb thrust for 30 seconds. This nozzle, *Fig.* 4, was applied to all JATO engines.

But the nozzle was to prove unsatisfactory. Failures began occurring at the thin section of carbon just downstream of the throat. The theory of the conical insert design had been that operating pressure would force the insert into the steel cone and the insert material would then be thrown into compression, under which stress it was quite strong. Probably because of thermal expansion of the steel and variations in cone angles in some cases, the carbon chipped away at the aft edge. This failure in many cases did not affect the performance of the engine, but occasionally the breaking of the carbon would proceed forward of the throat causing a drop in the chamber pressure and, consequently, in thrust.

These failures were eliminated by development of the cylindrical insert nozzle now in use, *Fig.* 5. In this design the thin carbon section at the aft edge of the insert is eliminated. Originally, cylindrical insert nozzles were made with a positive allowance between the outside diameter of the insert and the inside diameter of the steel body. These inserts cracked during firing tests because of the low tensile strength of carbon. Subsequent parts were made with a shrink fit between nozzle insert and body. This design gave excellent results and, after an extensive proof-test program, was incorporated into the unit.

Other features of this cylindrical insert design are the thin copper diaphragm held in place by the car-

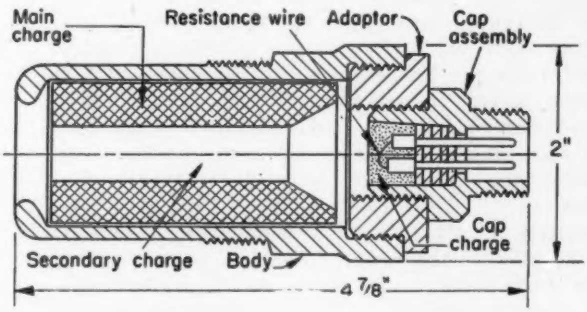


Fig. 6—Right—The first JATO igniter, patterned after an early Galcit design. Ignited by a resistance wire, the cap charge ignited the tubular pellet which, in turn, ignited the main charge by flame jet. Reproducibility was poor, lags of several seconds were common, and misfires and low-pressure burnouts occured frequently

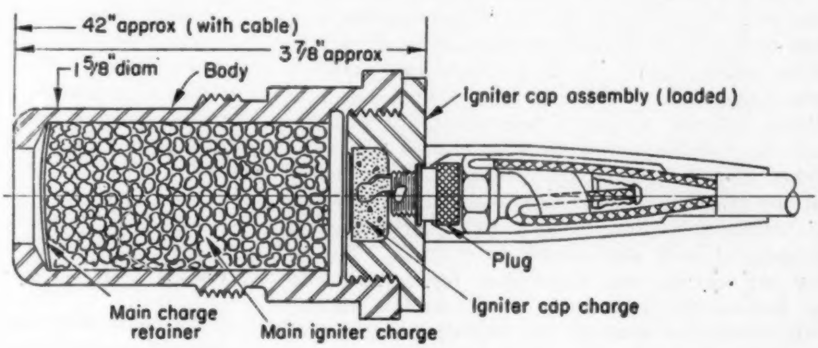


Fig. 7—Right—Ignition procedure for this improved igniter was the same as in the Fig. 6 igniter. A bag of cracked pellets was substituted for the tubular pellet primarily to provide both heat and pressure in the chamber—a deficiency of the flame jet principle of the previous igniter. Reproducibility was still difficult to maintain

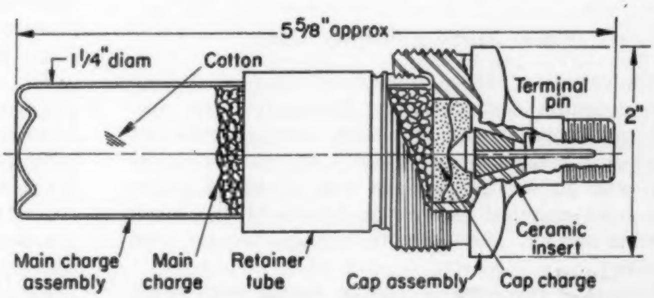


Fig. 8—Right—Present igniter design. Charge materials were changed to black blasting powder so packed as to overcome several earlier defects of ignition. It can be noted from this series of illustrations that the design of the igniter cap assembly was based on that of the familiar spark plug

bon insert and the protective cover over the exhaust end of the nozzle. The copper diaphragm insures reproducible ignition by maintaining chamber closure until the igniter is fired and the pressure inside the chamber has risen to about 500 psi. The mass of this diaphragm is not great enough to cause injury to the bystander when the engine is fired. The cap over the end of the nozzle protects the ignition diaphragm from damage during shipment and handling, and is removed just prior to firing.

IGNITER ASSEMBLY: The main problems in the development of the JATO igniter fall into two classes: charge development and mechanical development.

Obtaining Ignition Reproducibility

The first JATO igniter, *Fig.* 6, was patterned after the igniter used in an early Galcit design. A resistance wire was buried in a 5-gram charge consisting of black powder, aluminum dust, potassium perchlorate, and resin. This cap charge ignited a tubular pellet of Rezolin plastic mixed with potassium perchlorate, which in turn ignited the main charge. Incidentally, this pellet had the same form as many unrestricted-burning grains used in ordnance rockets. Design of the igniter was based on the assumption that the jet of flame from the ignition pellet, impinging upon the surface of the propellant charge, would raise the temperature of the charge to the ignition point. It soon became apparent that this igniter was inadequate. Reproducibility of ignition was poor, lags of several seconds were common, and misfires and low-pressure burnouts frequently occurred.

Analysis of the problem indicated that both heat and pressure in the chamber would be more effective than the flame jet of the plastic pellet. Accordingly, the plastic-potassium perchlorate mixture was produced in square pellets with 1/8 to 1/4-inch sides, and a bag of this material was substituted for the single tubular pellet, *Fig.* 7. Ignition was in every respect better than that of the previous design, but still ignition failures occurred. Reproducibility was difficult to maintain, even with a routine check on batches of pellets and tests on each batch to insure that the pressure reached in a closed chamber would fall within prescribed limits. Aging properties of the material were also subject to question. The igniter cap charge was responsible for several misfires because the mixture in the immediate vicinity of the resistance wire did not contain enough black powder to start the ignition sequence.

Black Blasting Powder Used

Elimination of these difficulties resulted in the present igniter design *Fig.* 8. The igniter cap mixture was replaced with fine black powder (FFF-G), which was poured into the cavity around the resistance wire and sealed in place with a disk of paper. The pellet material was replaced with (FFF) black blasting powder. Results with this igniter have been excellent.

While the problems of charge design were being solved, the mechanical features of the igniter were being perfected. In this phase of design, the main problem was the introduction of an insulated electrical circuit through a wall that necessarily had to be sealed against temperature and pressure.

The original igniter cap design *Fig.* 6 consisted of two pins pressed into a transite insert which was, in turn, pressed into a steel cap. The pins were arranged to mate with a standard socket insert AN plug. At the cap charge end of the insert the nichrome resistance wire was soldered between the pins. Failures occurring in this design sometimes involved the blowing out of the entire transite insert. Actually, the problems paralleled those of the design of the familiar spark plug, and that device formed the basis for the next igniter cap design, *Fig.* 7. The nichrome wire was soldered between the electrodes of the plug, and electrical leads soldered to the terminal and to a ground on the outside of the plug. Although this design solved the problem of the pressure and temperature seals, the arrangement of leads and of the fragile porcelain shank of the plug was not satisfactory. The present single-pole igniter, *Fig.* 8, was then proposed and developed.

1000 psi Gas-Leak Test Insures Operation

In this design a single positive lead wire is used, and the resistance wire is mounted in two parallel legs from the positive electrode to the ground. Originally a separate ground wire was planned, but later it was found that adequate grounding was provided at the contact between the JATO unit and the airplane mounting lugs. The single terminal enters the JATO engine through a conical ceramic insert which is cemented to the terminal and to the steel igniter body. A careful assembly procedure, which includes a 1000 psi gas-leak test, is necessary to insure successful operation of this design.

Design development problems involving the other components of the JATO rocket engine were similarly approached. The result is a device which provides a reliable source of power for assist take-off of aircraft, requiring only attachment to the airplane and an electrical impulse to initiate its operation.

They Say...

"In these days when there is general recognition that the human being is more important than the machine and that human power is more important than "natural" power, there is much that engineers can learn from the social scientists. Engineers must utilize the resources that come from learning from and working with those who know most of the ways in which the human body, mind and spirit function. Although it is true that the engineer has not often the technical training in the sciences that concern themselves with the human being, he is learning something of their content and also learning to cooperate with the sociologists, psychologists, psychiatrists and the others who work in these fields"—DR. LILLIAN M. GILBRETH, *president, Gilbreth Inc.*

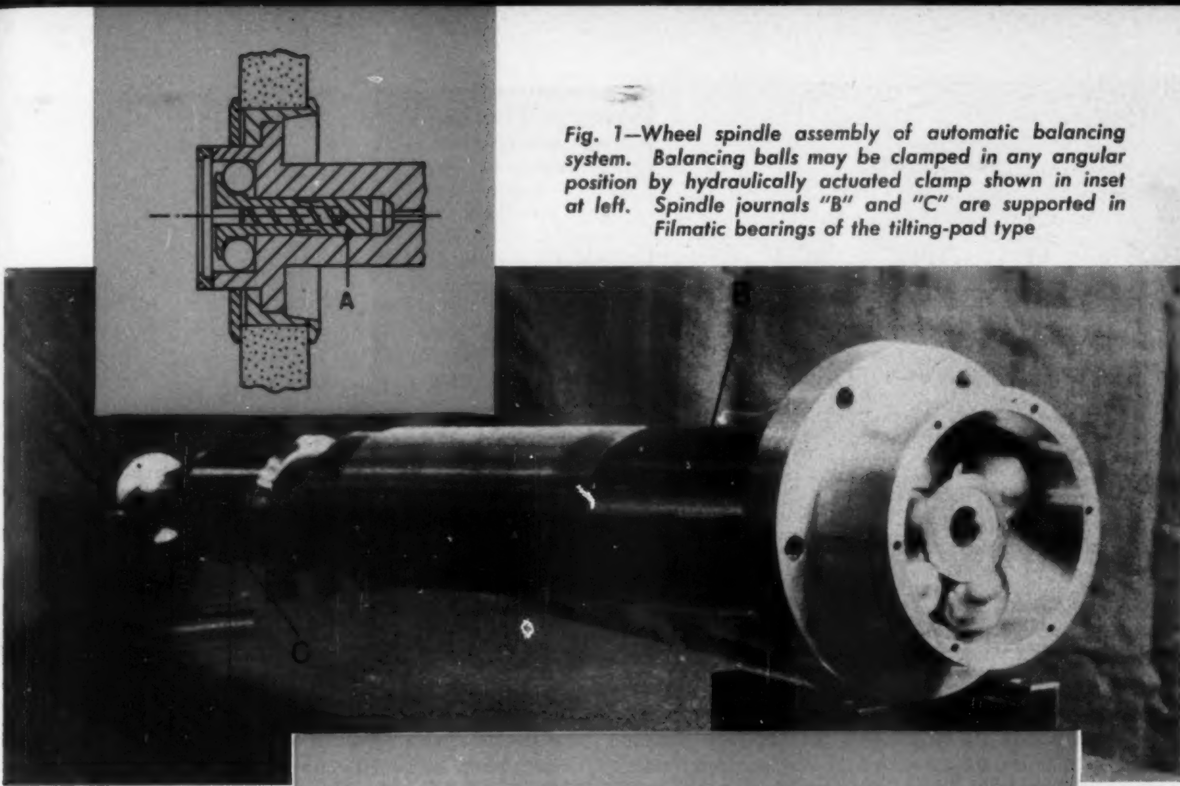


Fig. 1—Wheel spindle assembly of automatic balancing system. Balancing balls may be clamped in any angular position by hydraulically actuated clamp shown in inset at left. Spindle journals "B" and "C" are supported in Filmatic bearings of the tilting-pad type

Automatic Precision Balancing

. . . Speculation or experiment? Tests prove three-ball balancer does the job better than predicted by cursory analysis

By Hans Ernst
Research Director
Cincinnati Milling Machine Co.
Cincinnati, Ohio

BACK in the middle ages there was a long and learned debate concerning the number of angels that could stand on the point of a needle. Because the angels were immaterial beings, they could occupy no physical space; hence, the needle point could accommodate an unlimited number—so ran one side of the argument. But, said the opposition, the needle point itself has no physical dimension! Unfortunately, the argument was never settled because no one was ever able to induce the angels to stand up and be counted.

Later, there was another learned controversy about the speed of falling bodies. According to Aristotle and others, heavy bodies must fall faster than light bodies. This time the question was decisively settled by a practical-minded man named Galileo, who merely took similarly shaped heavy and light weights to the top of the tower at Pisa and dropped them. Their simultaneous crash swept away forever the speculative method as a tool of science and established the method of physical experiment.

So today, in seeking the answer to a physical problem or the workability of a complex new mechanism, we waste little time in speculative reasoning but proceed instead to apply the Galilean method of experiment and measurement: First, see what happens; measure the physical result!

AUTOMATIC BALANCING: Developed as an integral part of Cincinnati grinding machines, the automatic balancing system discussed in this article is an ex-

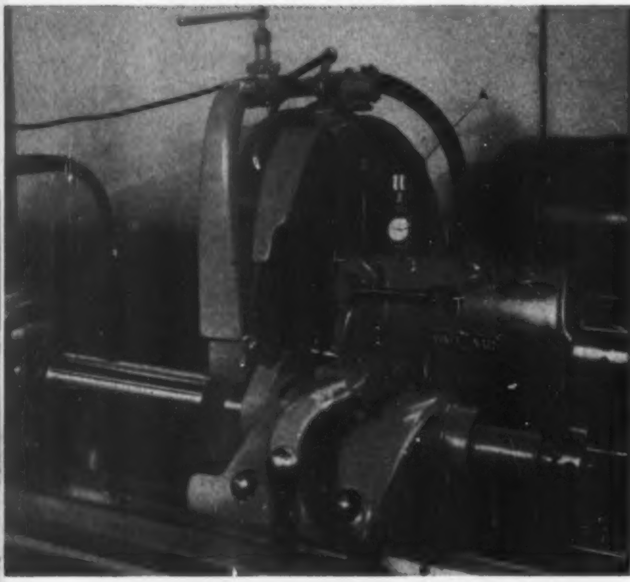


Fig. 2—Left—Balancing control system. Three-position lever "L" actuates clamping for both the bearing assembly, which is permitted to gyrate during balancing, and the balancing balls. Indicator "I" shows amplitude of vibration

Fig. 3 — Right — Test setup for determining accuracy of balancing device. Voltmeter "C" and oscilloscope "D" show amplitude of vibration picked up by crystal "A", amplified by "B"

ample of a physical system whose examination by the Aristotelian method might well lead to false conclusions. Its mode of operation is not obvious; many semingly impossible requirements must be met in order to arrive at a practical and reliable mechanism. First, let's see what happens. Does it really work?

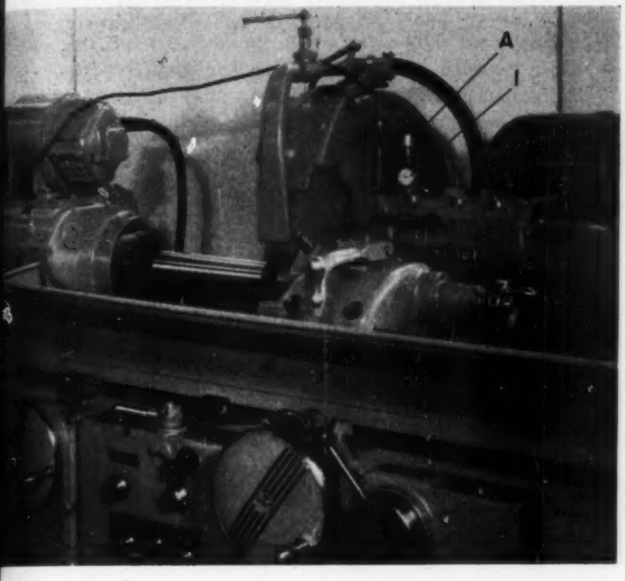
GENERAL DESCRIPTION: The purpose of this system is to provide a simple, automatic method of balancing a grinding wheel on the machine, eliminating the need of removing it and making tedious trial-and-error weight adjustments on a separate balancing stand. The balancing members consist of three steel balls carried in a raceway formed in the enlarged head of the grinding wheel spindle, as shown in *Fig.* 1. The balls may be clamped to the spindle in any angular position by the hydraulically actuated clamp "A". The spindle is carried in Filmatic bearings which comprise rigidly supported shoes of the tilting-pad type surrounding the journals "B" and "C." Wedge-shaped oil films developed by rotation between the journals and bearing shoes provide full hydrodynamic lubrication and act, in effect, as fluid vise-jaws to hold the spindle rigidly in position for precision grinding, while maintaining low friction and consequent freedom from wear. The entire front bearing assembly is mounted in a sleeve which may be either rigidly clamped to the wheel housing, as for the grinding operation, or unclamped and allowed to float in a special suspension of predetermined rigidity for the balancing operation. A three-position lever, "L," shown in *Fig.* 2, actuates both the bearing assembly clamping system and the ball clamp, independently. (In some of the production models, separate actuating members are used for the two clamping systems.) The rear bearing assembly is arranged to permit the slight angular deflections required during the balancing operation. An indicator, "I" in *Fig.* 2, shows when a balanced condition has been obtained and therefore tells when the system should be clamped.

INVESTIGATION OF ACCURACY: The machine chosen for this investigation was the 10-inch by 72-inch plain self-contained grinder shown in *Fig.* 3. This machine had been in service in a production operation for about one year, and in experimental and demonstration service for over three years. In this entire time the spindle bearings had not been adjusted or serviced, although the balancing device had been operated literally thousands of times.

To measure precisely the effectiveness of the balancing operation after this period of service, electronic vibration measuring instruments were set up as shown. A crystal-type displacement pick-up, "A," was mounted on top of the dial indicator, "I." This indicator shows the amount of unbalance by measuring the motion of the spindle suspension system when this is unclamped for balancing. The output from the pick-up was led to an amplifier, "B," and thence to a vacuum-tube voltmeter, "C," and cathode-ray oscillograph, "D." In each test, the amplitude of vibration of the spindle system was read from the vacuum-tube voltmeter and checked roughly with the reading of the indicator. The gain control on the amplifier was adjusted so that a motion of 0.002-inch on the indicator corresponded to a full-scale reading, 1 volt, on the vacuum-tube voltmeter. This provided an extremely sensitive means for measuring the motion of the spindle system in the unclamped condition. With this sensitivity it was found that, even when the spindle was stopped, the voltmeter showed a background reading of 0.10-volt because of extraneous vibration. This background indication, therefore, was subtracted from all readings taken in the tests made with the spindle running.

Three series of balancing tests were made, differing primarily in the extent and location of the initial unbalance.

Series 1: The balancing operation was started for each test with both an added weight and the balls located at the bottom. Under this condition an unbalance of about 44 ounce-inches existed in the machine



Series 2: These tests were essentially the same except that the added weight was relocated 180 degrees from its position in the first series. Therefore, when the system was balanced, the balls were on the opposite side of the runway from the position they occupied in the first series

Series 3: For these tests several small weights were so located that the three balls would be positioned almost exactly 120 degrees apart after the balancing operation. This indicated that the system was almost perfectly balanced without the balls. Each test in this series was started with the balls at the bottom, an unbalance of about 24 ounce-inches.

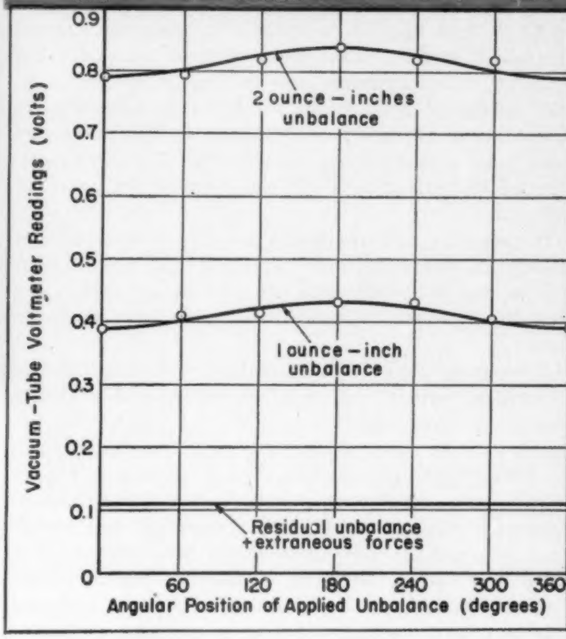
Ten separate tests were made by each of two operators in the first and the second series and five tests in the third series. The voltmeter readings for the 50 tests are given in TABLE 1.

CALIBRATION: A series of tests was then carried out with the double purpose of, first, calibrating the entire system so that the vacuum-tube voltmeter readings could be properly interpreted in terms of unbalance, and second, determining as accurately as possible the actual values of residual unbalance.

With no added weights, the grinding wheel was first balanced by releasing and clamping the balls in the normal way, and an average value of residual unbalance obtained (as compared with the readings in 50 tests). The reading of the vacuum-tube voltmeter corrected for static background (with the spindle bearing suspension system unclamped) was found to be approximately 0.11-volt. This position of the balls was then maintained throughout this series of tests.

A weight corresponding to 1 ounce-inch of unbalance was then applied to the grinding wheel mount at a selected position; the spindle was rotated at normal speed, and a voltmeter reading taken. The weight was then moved successively to positions spaced 60 degrees apart around the wheel mount, at the same radius, and a voltmeter reading taken for each position. The suspension system, of course, re-

Fig. 4—Below—Results of calibration test. Slight residual unbalance caused cyclic effect which permitted division of the constant ordinate (0.11-volt) into two components: actual residual unbalance (0.03-volt) and extraneous dynamic effects (0.08-volt)



mained unclamped throughout these tests.

The tests were then repeated with a weight corresponding to 2 ounce-inches of unbalance. The readings for both groups of tests (corrected for static background) are shown plotted in *Fig.* 4.

Inspection of *Fig.* 4 reveals that the straight line corresponding to a constant ordinate of 0.11-volt for no applied unbalance includes not only the effect of the residual unbalance due to improper positioning of the balls, but also the effect of other dynamic forces such as those caused by slight unbalance in the driving motor and belts and magnetic pulsations in the motor. These forces come into play only when the spindle is running and hence are not included in

Table 1—Voltmeter Readings for Balacing Tests*

Test No.	Series 1 Operator A	Series 1 Operator B	Series 2 Operator A	Series 2 Operator B	Series 3 Operator A	Series 3 Operator B
1	0.14	0.12	0.10	0.12	0.10	0.16
2	0.14	0.12	0.10	0.14	0.14	0.12
3	0.16	0.12	0.12	0.16	0.18	0.14
4	0.10	0.10	0.16	0.10	0.12	0.10
5	0.12	0.10	0.12	0.14	0.10	0.12
6	0.14	0.08	0.08	0.12		
7	0.18	0.22	0.10	0.14		
8	0.10	0.18	0.10	0.16		
9	0.14	0.08	0.14	0.14		
10	0.16	0.10	0.12	0.12		
Average:	0.138	0.122	0.114	0.134	0.128	0.128

* Readings corrected for background effect obtained with spindle stopped. Average corrected reading for 50 tests = 0.127 volt = 0.17 ounce-inch residual unbalance.

the "static" background for which a correction can easily be made.

The lines through both sets of plotted points show a slight cyclic rise and fall. This, of course, is to be expected, inasmuch as any unbalance remaining from the balancing operation would be vectorially added to (or subtracted from) the effect of the test weights as they were applied to successive positions around the wheel mount. Taking one-half of the difference between the maximum and minimum ordinates in each group of tests therefore gives an approximate indication of the residual unbalance in this particular case. As shown by the curves, *Fig.* 4, the half-difference in both groups of tests was of the order of 0.03-volt.

It therefore appears that the effect of the residual unbalance was only about 0.03-volt and the remainder of the 0.11-volt ordinate (or about 0.08 volt) was due to other dynamic forces such as those listed before.

Averaging the voltage ordinates for each of the two groups of tests therefore gives values which should provide a good calibration for the entire system. These values, together with the value of approximately 0.08-volt for zero unbalance, are plotted in *Fig.* 5.

Inspection of *Fig.* 5 shows that the relationship between vacuum-tube voltmeter readings and unbalance is generally linear, as would be expected. Toward the origin, there is a small departure from linearity which may be caused by the slight damping inherent in the spindle bearing suspension system.

Results: TABLE 1 shows that the average of the vacuum-tube voltmeter readings for the entire 50 balancing tests was 0.127-volt. Referred to *Fig.* 5, this shows an average unbalance of 0.17 ounce-inch. The highest reading in all these tests was 0.22-volt, which corresponds to 0.47 ounce-inch. The lowest reading was 0.08-volt, which is actually the value shown by the calibration curve for zero unbalance. This, of course, may still include a small experimental error; however, it indicates that in some cases almost perfect balance was obtained.

In *Fig.* 6 is shown a statistical distribution curve for the 50 readings listed in TABLE 1. Only one reading was as high as 0.22-volt (0.47 ounce-inch). Even an occasional value as high as this, however, has little real significance since in practice this could be further reduced if desired. A slightly longer period could be allowed for the balls to assume a more favorable position, as shown by a smaller motion on the indicator, or the spindle could be stopped, the balls dropped, and the balancing operation repeated with a new initial configuration of the balls.

In view of the results obtained in these tests, it may be stated conservatively that this automatic balancing device will balance the grinding wheel system to less than ½ ounce-inch of error.

Grinding Test: Several questions may logically be asked: Is a residual unbalance of ½ ounce-inch sufficiently small to permit a high-precision grinding operation? How does this value compare with that usually obtained by skilled operators using the old method of removing the grinding wheel from the machine and balancing it on a balancing stand?

The first question was answered by actual grinding tests. The wheel system on the machine was first carefully balanced until the vacuum-tube voltmeter showed a reading (when corrected for background) that corresponded to a virtually perfect balance. A grinding operation was then performed on the workpiece shown in position in *Figs.* 2 and 3. With a 60-grit, vitrified-bond, aluminum-oxide wheel,

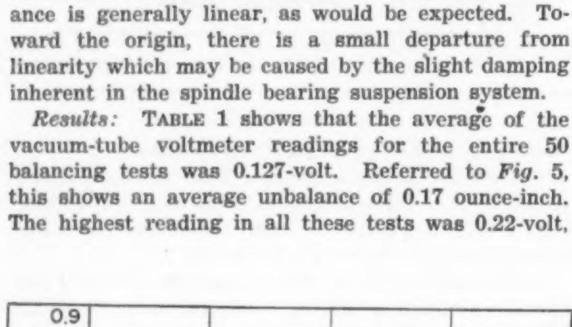
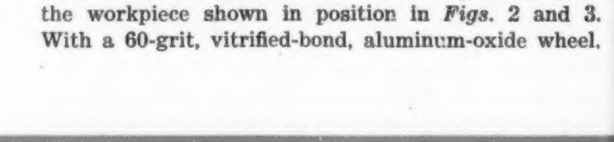


Fig. 5—Left—Calibration curve relating voltmeter readings to actual unbalance developed from data of Fig. 4

Fig. 6—Below—Distribution curve showing results of 50 balancing tests



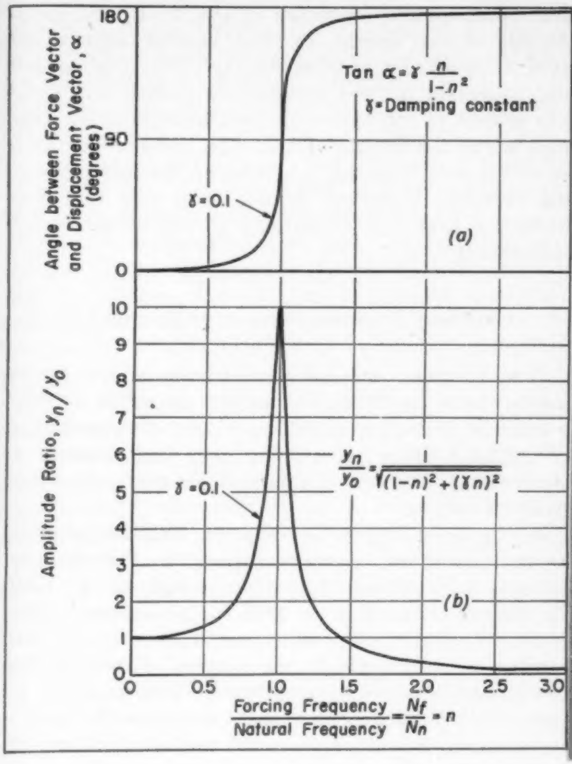


Fig. 7—Left—Characteristics of vibration inherent in balancing system

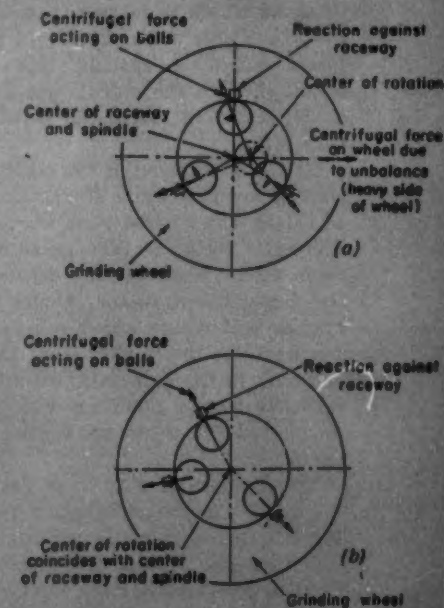


Fig. 8—Right—Force systems for balls in unbalanced state above resonance (a) and balanced (b)

the surface finish obtained on the SAE 3145 steel workpiece (375 Bhn) averaged about 3 microinches and the bar was round within 0.000,01-inch.

A weight corresponding to ½ ounce-inch of unbalance was then attached to the grinding wheel mount, and the grinding operation repeated. There was no discernible difference in the quality of grind obtained. The finish again averaged about 3 microinches and the bar was round within 0.000,01-inch. This test showed conclusively, therefore, that a residual unbalance of the order provided by the automatic balancing system was small enough to permit a high-precision grinding operation.

The second question also was answered by test. Several grinding wheels were first balanced on a balancing stand by a skilled operator. The wheels were then carefully checked for residual unbalance on a Gisholt Dynetric balancing machine. The *average* value of this unbalance was found to be on the order of one ounce-inch, which is more than twice as much as the highest residual unbalance found in the described tests with the automatic balancing system.

THEORY OF OPERATION: A positive answer has been furnished to the question: Does it work? There then arises the question: How and why does the three-ball balancer work?

It was shown by Thearle* in 1932 that an automatic balancing mechanism could be provided for a rotating system by using two balls mounted in a raceway concentric with the axis of rotation and running the system somewhat above resonance. The physical principle embodied in this mechanism is based on certain aspects of the theory of vibration of elastic bodies.

In accordance with vibration theory, the phase angle between force and displacement in an elastic system excited by a sinusoidally varying force is a function of the ratio of the forcing frequency, N_f, to the natural frequency of the system, N_n. When this ratio is zero, the phase angle is likewise zero. When the ratio is unity, which is the condition of resonance, the phase angle is 90 degrees, and when the ratio is infinity, the phase angle is 180 degrees. Intermediate values depend on the amount of damping present in the system. As shown in *Fig. 7a*, for a system containing little damping the phase angle remains small while the ratio N_f/N_n increases from zero to almost unity, then rises abruptly, passing through 90 degrees when $N_f/N_n = 1$, and closely approaches 180 degrees for values only slightly higher than 1. As shown in *Fig. 7b*, when the exciting force is of constant amplitude, the vibration amplitude peaks sharply when the ratio is unity, and falls rapidly for higher values.

In a rotating system any unbalance weight, *W*, provides a rotating force vector, of magnitude $Wr\omega^2/g$, which acts with respect to any given radial direction as a sinusoidally varying force. If the system is so suspended as to have equal rigidity in all radial directions, the resultant orbital motion is therefore circular. If the rigidity, *K*, of the suspension and the mass, *M*, of the rotating system are such that $\omega^2 < K/M$, where ω is the angular velocity in radians per second, the system runs below its resonant speed. The ratio N_f/N_n is therefore lower than unity, and

*E. L. Thearle—"A New Type of Dynamic Balancing Machine," *Trans. ASME*, Vol. 54, 1932, Pages 131-141.

the force vector is only slightly out of phase with the displacement vector. This means that under these conditions, the entire system will rotate about a point in space which is remote from its "heavy" side, i.e., the *heavy* side will run *out*. In the case of the automatic balancing system, this corresponds to the condition with the spindle assembly rigidly clamped, as for grinding. Here the natural frequency, N_n, is very high.

On the other hand, if the rigidity, K, of the suspension is lowered while M remains the same, so that $\omega^2 > K/M$, the system runs above its resonant speed. The ratio N_f/N_n is then greater than unity and the force vector is almost 180 degrees out of phase with the displacement vector. Under these conditions, the system will rotate about a point in space which is remote from the "light" side; i.e., the *light* side will run *out*. In the automatic balancing system, this corresponds to the condition with the spindle assembly unclamped and supported resiliently for the balancing operation.

The diagram in *Fig.* 8*a* shows the force system of the 3-ball Cincinnati balancer when the spindle bearing suspension is in the unclamped position. In this condition the system is operating above resonance. If at a given instant the heavy side of the grinding wheel system is at the right, the unbalance force vector will act in the direction of the arrow while the displacement vector will be to the left. The system will therefore rotate about a point in space which is toward the heavy side of the wheel.

Balls Forced to Light Side

As shown by the arrows, the centrifugal force on each ball acts in a radial line from the actual center of rotation. Because of the eccentricity of the raceway, this force does not pass through the point of contact of ball and raceway, and therefore produces a couple which urges each ball toward the light side of the system.

If the balls are unclamped, they will migrate toward the light side. In doing so, they reduce the unbalance in the system; consequently, the center of rotation approaches the geometric center of the raceway. When the balls have reached positions where the system is completely balanced, the two centers coincide, the centrifugal force on each ball is in line with the reaction from the raceway, and thus there is no longer any couple tending to move it. This condition is shown in *Fig.* 8*b*. Then the balls are clamped; next, the bearing system is clamped, thus providing the highly rigid condition required for grinding.

Although the mechanical elements and operating principle of this balancing system are simple, several difficult and conflicting requirements must be met in order to provide a practical mechanism and obtain a degree of balance satisfactory for the needs of a precision grinding machine.

Concentricity: From *Fig.* 8*b* it is obvious that if the raceway containing the balls is not exactly concentric with the journal axis, a perfectly balanced condition cannot be obtained. For this reason, the raceway is formed as an integral part of the grinding wheel spindle, as shown in *Fig.* 1. In the manufacture of this spindle, extreme care is taken to insure a high order of concentricity. The finish grinding operation on the raceway is carried out while the spindle is rotated on its own superfinished journals supported in special bearings, and care is taken to obtain a high quality of finish on the ball supporting surface. It seems obvious that once this concentricity has been established it will be inherently maintained.

Friction Prevents Accurate Balance?

Low friction: Another requirement which would appear to be essential for proper operation of this automatic balancing system is a very low coefficient of rolling friction between the balls and raceway. A conventional analysis of this problem would be somewhat as follows:

In the force diagram of *Fig.* 9*a*, which represents an unclamped and unbalanced condition, F is the centrifugal force vector acting on a ball, at A, from the center of rotation O. This force produces a moment, $F(r/R)\ e \cos\theta$, which tends to move the ball toward the light side of the system. Here e is the eccentricity caused by unbalance. But in order to move the ball, this moment must overcome the opposing moment, Ff_r, where f_r is the coefficient of rolling friction.

Therefore, $(r/R)\ e \cos\theta$ must be larger than f_r, i.e., $e \cos\theta$ must be larger than $f_r\ (R/r)$. If $R/r=$ 2.8, as in the spindle assembly shown in *Fig.* 1, then even for the position A', where the moment tending to cause motion of the ball is a maximum (because θ is zero), e must be larger than 2.8 f_r.

If it is assumed that the coefficient of rolling friction, f_r, has a value of about 0.0005-inch such as commonly experienced for conditions similar to those found in the balancer, the minimum eccentricity of the raceway during the balancing operation, beyond which the balls would not be expected to reposition themselves from any location, would be (2.8) (0.0005) = 0.0014-inch.

If this were true, the residual unbalance in the system could indeed be objectionable. With the minimum eccentricity during balancing equal to 0.0014-inch, the distance between the center of mass and the axis of the journal would be expected to be about two-thirds of this amount, or 0.0009-inch. When running above resonance, the total eccentricty, e, of the journal axis is equal to the eccentricity of this axis with respect to the mass center plus the eccentricity of the mass center itself. For a value of N_f/N_n such as used in the balancing device, the eccentricity of the mass center is approximately 0.33e. The effective weight of the rotating system for a 30-inch by 3-inch grinding wheel being 300 pounds, the residual unbalance would be (300) (0.0009) = 0.27 pound-inch or 4.32 ounce-inches.

If a ball were situated in a less favorable location than A' in *Fig.* 9*a*—for example, at the position A, or any of the positions shown in *Fig.* 8*a*—the eccentricity required to produce motion would be even higher. In any of these positions the angle θ has an

appreciable value; therefore, e must have even a larger value than in the preceding example, and the residual unbalance should be even greater.

In view of the foregoing analysis, it would appear to be impossible to achieve a high degree of balance with this type of system. How then is it possible to balance the grinding wheel system consistently to a small fraction of an inch-ounce?

The answer lies in the fact that a static condition, assumed in the foregoing analyses, is not the actual case. In reality, there exists a dynamic condition with the ball being acted upon by several sets of forces which may combine to produce a resultant motion in a desired direction.

It is well known that a shaft rotating in a bearing may be moved endwise by an extremely small force. Here, the actual motion opposed by the friction in the bearing is the resultant of the tangential and axial motions; consequently, when the axial velocity is very small in comparison with the tangential velocity, the axial force component required to produce this low velocity is likewise very small.

Similarly, a ball rolling in a straight line on a flat surface under the action of a single force is diverted from its straight path when even a minute force is applied in a perpendicular direction. The actual motion of the ball is in the direction of the resultant of the two forces. The same principle applies in the case of a ball lying on a surface and subjected to an oscillating tangential force to which is added a small colinear force of constant direction and magnitude. If the moment produced on the ball by the oscillating force has a peak value higher than the friction moment, an oscillatory motion occurs. The acceleration produced by the small constant force then alternately increases and decreases the velocity of the ball, the resultant motion being a progressive advance in the direction determined by the constant force.

This is the state of affairs in the ball-type balancing device. As indicated in *Fig. 9b*, each ball is urged toward the raceway by a centrifugal force vector, $F = W\omega^2 R/g$. With the dimensions and velocities used in a typical application, F is approximately equal to $40W$. Then, if the coefficient of rolling friction, f_r, were 0.0005-inch, the friction moment opposing the rolling of the ball would be $F f_r = 0.02W$. In addition to centrifugal force, however, a constant downward force, W, acts on the ball due to gravity and has a tangential component, $W \sin \omega t$. This component constitutes an oscillating force which alternately acts to accelerate and retard the ball once every revolution of the spindle. At a peak value, as when the ball is at A where $\sin \omega t = 1$, the tangential component is equal to W. Thus for a ball radius, r, of 0.687-inch there is produced a rolling moment equal to 0.687 W, which is far greater than the opposing friction moment, $0.02W$. The ball will therefore oscillate once every revolution through a small angle in its raceway in a plane perpendicular to the spindle axis.

Oscillatory Action Overcomes Friction

The approximate amplitude of oscillation of the ball in its raceway has been computed. In this example, the total excursion of each ball forward and back, once every revolution of the spindle, is approximately 0.080-inch, a significant quantity.

For the slightest amount of unbalance in the entire grinding wheel system, there is a small eccentricity of the raceway with respect to the center of rotation, which causes a small constant force on each ball urging it toward the light side of the system. As described, even if this force is extremely minute it still produces motion in the desired direction when combined with the oscillating force, because this in itself is more than large enough to overcome the frictional opposition. Thus the balls, in effect, are unhindered by rolling friction and consequently can migrate to the positions where the system is perfectly balanced. This is in accordance with the observed result.

Other dynamic factors that may act during the balancing operation to produce slight oscillatory motion of the balls in their raceway, and consequently aid in their perfect positioning, include small departures from constant angular velocity of the revolving system due to slight departures from a perfectly circular orbit. Any small cyclic changes in radius of curvature, as in a slightly elliptical orbit, would introduce significant accelerations and retardations of the balls with respect to the raceway.

The apparent absence of any effective frictional opposition to the motion of the balls while they are seeking their balanced positions is further evidenced

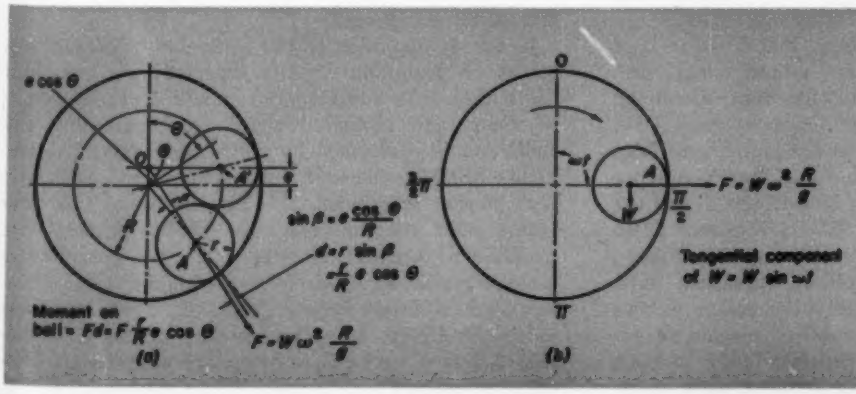


Fig. 9—At (a), force diagram for ball including friction effect. At (b), diagram showing centrifugal force. Tangential component of weight produces oscillatory action that overrides frictional resistance

by the fact that it is necessary to introduce a small amount of viscous damping, or intermittent clamping, to prevent the balls from hunting after their balanced positions have been reached.

Bearing Design: An important factor in the successful application of this balancing principle to precision grinding machines has been the particular construction of the bearing system.

Bearings Withstand Unbalance Shocks

Fears have been expressed by some that the bearings might be damaged, or their adjustment disturbed, by the vibrations or impacts imposed by an occasional heavy unbalance during the balancing operation. Obviously, this might be true if the bearings were inherently fragile in nature, or required frequent microsensitive adjustment. However, with the preloaded oil-film type of bearing, as used in all the machines with the three-ball balancer, no such difficulty has been experienced. These bearings are inherently rugged and rigid. As indicated earlier, the machine that was tested for balancing accuracy had been subjected to severe production service and the balancing system had been actuated thousands of times. Yet the machine performed a grinding operation of high precision in spite of the fact that the bearings had never been readjusted since their initial assembly more than four years before. Similar performance characteristics have been shown in the field by other machines of this type.

CONCLUSION: Throughout the ages, whenever new mechanical devices were proposed or made, there were always people who said, "It can't be done!" They were not always ignorant people—often they were highly intelligent and highly trained. They could prove their contentions mathematically. They were not wrong because of their theories, they were simply not in possession of all the facts; their assumptions were untenable.

The practicability of almost any new device appears questionable in the early stages of its development. There is always a long hard road to be traveled between the class-room demonstration of a physical principle and its application in a practical machine capable of meeting the multifarious requirements of modern production. Application of the three-ball (dynamically positioned mass) principle of automatic balancing to commercial centerless and center-type grinders has provided no exception to this general rule. Of course, there were difficulties. Masses and rigidities had to be properly proportioned; careful consideration had to be given to the details of the design so as to provide controlled damping and to ensure that the precise concentricities once established would be inherently maintained; the bearing system had to be designed so as to be unaffected by the forces produced by unbalance during the balancing operation—and by variations in belt pull.

Now, precise balancing is accomplished in seconds instead of an hour or two. The balancing operation can therefore be repeated daily, or as often as desired, without appreciable loss of time. Year-after-year operation of the entire bearing system has been accomplished with complete freedom from adjustment or maintenance. These facts prove the correctness of this balancing principle and the effectiveness of its application.

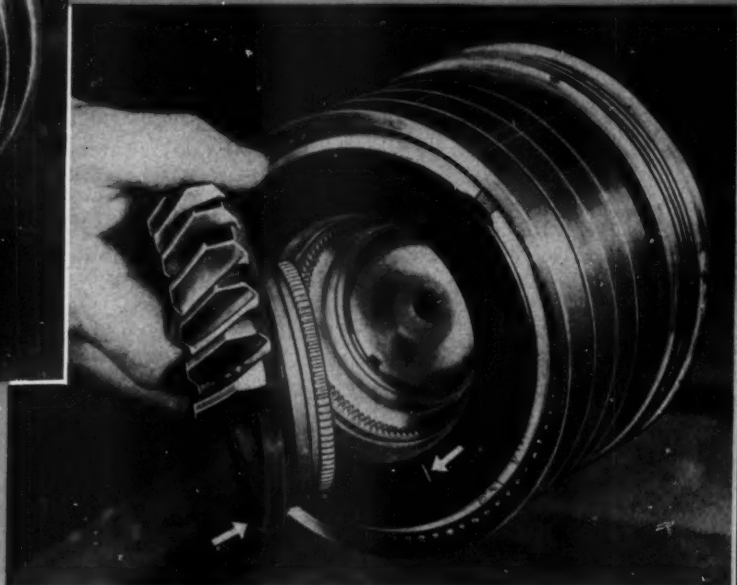
Computer Solves Military Problems

A HIGH-SPEED, general-purpose, automatically-sequenced electronic computer has been developed and constructed by the National Bureau of Standards under the sponsorship of the Department of the Air Force. It provides a high-speed computing service for Air Force project SCOOP (Scientific Computation of Optimum Programs), a pioneering effort in the application of scientific principles to the large-scale problems of military management and administration. The computer automatically performs all of the logical and arithmetical operations required to solve a particular problem when it is supplied with coded instructions and numerical data. All operations are carried on in a binary number system, the presence of a pulse indicating "1" and the absence of a pulse indicating "0." A sequence of 45 binary digits is known as a "word" and may convey operational instructions as well as numerical data. High computing speed is largely the result of two design features: rapid pulse rate (1 megacycle per second) and large memory capacity (512 words). They make it possible to add or subtract pairs of 11-digit numbers 1100 times a second and multiply or divide them 330 times a second. These rates include the time it takes for the machine to search its memory for the numbers, operate upon them and return the result to the memory. An arithmetical operation of addition or subtraction alone is completed in 50 microseconds; multiplication or division is completed in 2500 microseconds.

In the development of the computer, emphasis was placed on designing circuits especially for computer use rather than adopting the standard procedures of television and radar circuitry. All computing and switching is performed by germanium crystal diodes rather than by electron tubes; tubes are used only for power amplification. Standardized tube-and-transformer combinations are used throughout the machine to simplify maintenance. The transformer method provides high-frequency coupling with a minimum of cross-talk between circuits. These advances in circuit design, as well as several unique construction features, make it easy to add units to the machine to convert it to a much larger computer.

Fig. 1—Left—Aircraft propeller positive lock showing splined locking ring used for blade adjustment

Fig. 2—Below—Propeller hub with ring in place and co-ordinating gear being inserted



Splined Positive Lock

. . . provides fine feed screw nut adjustment in 0.00016-inch steps

By Thomas Barish
Consulting Engineer
Cleveland, Ohio

AIRCRAFT propellers require a positive lock with extremely fine adjustments for holding the blades against change of angle. The lock must be positive because of the large torque tending to force the blade to zero angle, and because of the severe vibration. The fine adjustment is necessary because the angle of the various blades must be equal to within 0.1-degree for aerodynamic balance.

The type of lock developed for these requirements is illustrated in *Figs.* 1 and 2. The intermediate locking ring, *Fig.* 1, has teeth on the outside that mesh with a similar spline on the blade; and teeth on the inside mesh with a corresponding set on the co-ordinating gear, *Fig.* 2, that holds all the blades in relation to each other. Each of these sets of gear teeth have all the teeth in mesh at once, providing great torque strength and permitting relatively light parts.

Adjustment is obtained because one of the two splines has one less tooth than the other. For example, if one set has 90 teeth and the second set 91 then there are 90×91 or 8190 adjustment positions in the circle. Rotating the locking ring one tooth will adjust the blade by 1/8190 part of a circle or 0.044-degree.

The same method has been applied to a machine tool feed screw to remove all end play to within about 0.0001-inch. In the construction shown in *Fig.* 3, the nut has an external spline with 27 teeth. The reverse thrust ring has an external spline with 23 teeth. The floating lock ring on the outside meshes with both of these splines and is held in place by a snap ring. Thus with 27 and 23-tooth gears, adjust-

ment can be obtained to within $1/27 \times 23$ part of a circle or 0.00016-inch with a ten-pitch screw. Also this adjustment can be made without taking off the nut or endplay collar.

However, there was difficulty in using this lock. When the two sets of spline teeth differ by more than one tooth, the minimum steps are not obtained by moving the locking collar one tooth. In fact, for this case moving the locking collar one tooth changed the endplay by 0.00064-inch or four times the expected 0.00016-inch. Also the steps are so small that one cannot pick them out by "feel", and trying all of the 621 possibilities is impractical. One must know how many teeth to move to obtain the desired minimum steps, so as to approach the "no looseness" point gradually.

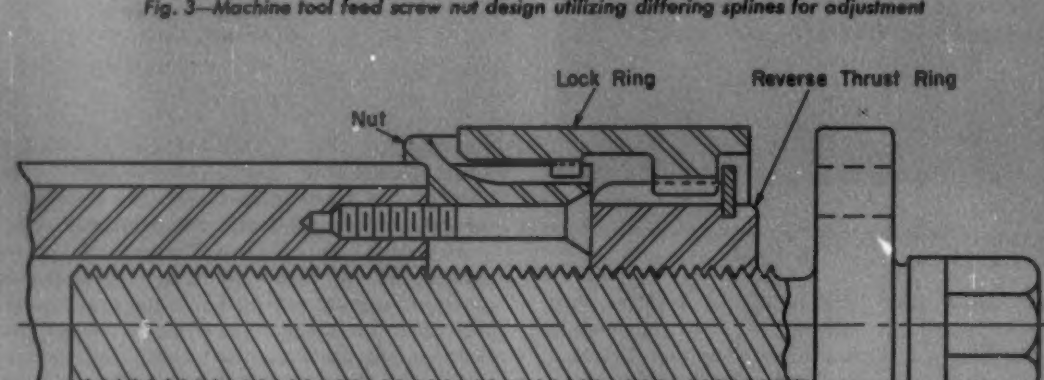
The formula for this is an interesting and simple problem in the mathematics of integral numbers: Let a = teeth in one spline and $a + d$ = teeth in the second. Thus the expected minimum adjustment is $1/a(a+d)$ part of a circle. Advancing the locking ring one tooth will make an adjustment of $1/a - 1/(a+d)$ or $d/a(a+d)$ part of a circle. If the difference d is one, then an advance of one tooth will give the expected minimum step. But if d is more than one, for example 2, then the step of one tooth will give $2/a(a+2)$, or about twice the minimum step, and will continue to give too large steps.

To get the minimum step the small spline must be advanced some number of teeth such as n, and the second, or larger spline, should be moved up one more, or $n+1$. Then to find n, set up the equation for the minimum expected step value, equal to the distance one spline advances less the distance the second spline returns: $1/a(a+d) = n/a - (n+1)/(a+d)$. Solving, $n = (a+1)/d$, where a is the number of teeth in the smaller spline, d is the difference between the two splines, and n is the advance for minimum steps. Thus for *Fig.* 3 where 27 and 23 teeth are used: $n = (23+1)/4$ or 6 teeth. And to get minimum steps the inner spline is advanced 6 teeth and the outer spline 7 teeth.

With these instructions, mechanics are able to handle the lock easily. After roughly approaching the desired adjustment, the 6 and 7-tooth steps are tried one at a time until the desired fit is obtained.

Note that such a design must have the numbers of teeth selected such that there are no common divisors or many of the adjustments will be duplicated. In the case of *Fig.* 3, it was necessary to have a 4-tooth difference so the bore of the larger spline would clear the OD of the smaller one. Note also that one spline is made much longer than the other so it enters first and helps materially in handling adjustments since the collar can be pulled back off the first spline and still hold on the other for the rough adjustment.

Fig. 3—Machine tool feed screw nut design utilizing differing splines for adjustment



Designing Taper Involute Splines

Proper analysis of these recently developed splines shows that the compression rather than shear stress is critical in calculating spline length

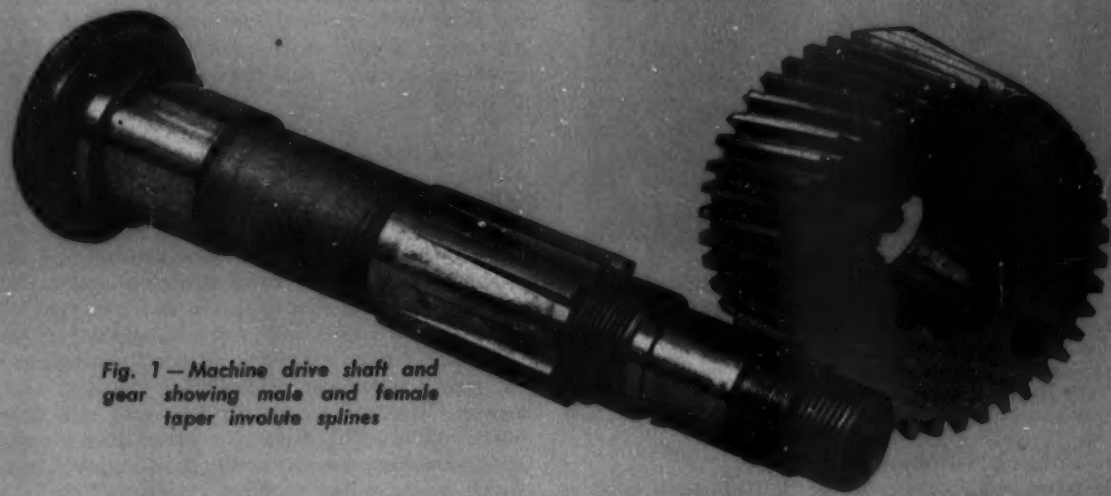


Fig. 1—Machine drive shaft and gear showing male and female taper involute splines

MEANS for holding machine parts together to prevent relative rotation has been improved greatly during the past few years. From the single, multiple and parallel-key spline, design has progressed to the standard involute spline. Similarly, the holding means on a taper bearing has advanced from a single key on a tapered shaft to the parallel-key, taper-root spline and now to the involute taper-root spline, *Fig.* 1.

For power transmission, the involute spline is rapidly gained popularity over the parallel-key spline. The successful application of the parallel-root involute spline has led to the adaptation of the involute form to the taper-root design. Several factors contribute to the success of both types of involute splines. Universal cutting tools within a diametral pitch, increased strength, freer machining of both members, more economical cutting tools, and longer life of the hob are some of the factors which are contributing to the changeover to the involute spline. The parallel-root involute spline has been successfully applied to many types of machine drives including automotive, aircraft, machine tool, tractor, farm implement, and industrial.

INVOLUTE TAPER-ROOT SPLINE: The involute taper-root spline has exceptional possibilities where increased strength and additional bearing are desired. Heavy power transmission may be improved by the use of this type of spline. Some parts to which taper involute splines have been successfully applied are tractor axles, machine tool drives, and automotive drives such as the steering sector and spindle, brake-lever shaft and pedal clutch driving disk, sliding drive gear and reduction gear, steering spindle and gear, and brake shaft and pinion.

Where it is desirable to take up any looseness due to wear, the involute taper-root spline can be successfully applied. As indicated by the list of applications, it can be used on all reversible drives excepting those for extremely heavy work. Drives in which the members must be easily disassembled can employ this type of spline very successfully. Another important feature of this spline is that it will run true because it is fitted onto a taper whereas any inaccuracies in other type of connections may result in the members not running true.

MACHINING SPLINES: The hob for producing a parallel-key spline is difficult to manufacture because the profile must be a curve rather than a straight rack as on an involute-spline hob. Both manufacture and inspection are complicated by this requirement. Inspection by the user of this type of hob is also more difficult.

One of the most important advantages of using the involute form is that a hob of a given pitch will cut a spline of any standard pitch diameter for that pitch, whereas a hob for a parallel-key spline will cut only the piece for which it was designed. This could amount to a considerable saving in tool cost where it is necessary to cut several different diam-

---

This article is based on information and data furnished by Mr. John P. Breuer, Hob Engineer, the Barber-Colman Co.

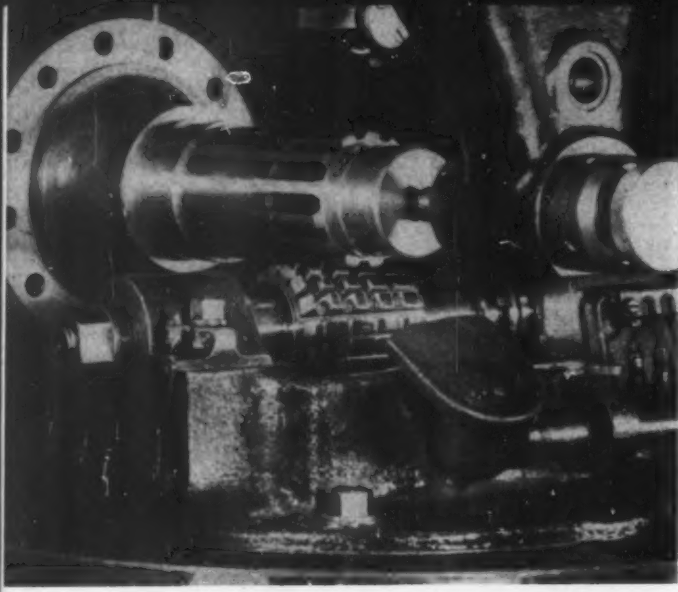


Fig. 2—The oblique feed of the hob slide on the Barber-Colman No. 16-11 hobbing machine is the distinguishing feature which allows it to cut taper-root splines. It can be seen that the long teeth of the hob have already cut the outer end of the spline to full depth while the short teeth will finish the keys at the inner end of the taper

eters of the same diametral pitch.

Of comparable importance is the fact that an involute-spline hob can be used to produce all three classes of fits on both major diameter and side bearing applications. The class of fit for side bearing is controlled by varying the depth of cut. A different hob is required for fillet root type and flat root types. A hob with a shorter depth of form is required for the seldom-used minor diameter bearing. The class of fit for a minor diameter bearing is controlled by varying the size of the hole in the internal splined member.

Many parallel-key splines require a hob with long

Types of Splines

Parallel-Key Spline: Straight-sided key with the sides of the key parallel and the root parallel to the axis.

Parallel-Key, Taper-Root Spline: Straight-sided key with the sides of the key parallel and the root at an angle with the axis.

Involute Spine: 30-degree involute form on the sides of the key and the root parallel to the axis. All basic dimensions are based on 1/2 diametral pitch. Tooth size is designated by a pair of numbers such as 1/2, 2/4, etc. Number of teeth divided by the first digit equals the pitch diameter; the reciprocal of the second digit equals the addendum or dedendum, which are equal.

Involute Taper-Root Spline: 30-degree involute form on the sides of the key and the root at an angle with the axis. All basic dimensions are based on 1/2 diametral pitch, as with the involute spline.

protuberances on each tooth to insure that the straight portion of the side of the key will extend to the required depth. These protuberances are subject to rapid wear and cutting hazards. Standard involute spline hobs do not have protuberances, which greatly increases the number of workpieces between sharpenings as well as the total life of the hob.

Special hobs with a tapered outside diameter are required to cut involute taper-root splines, *Fig.* 2. A single hob will cut any standard pitch diameter of the same pitch and any conventional included angle. All classes of fits can be cut with the same hob and, of course, a minor diameter bearing will be used in all cases.

Internal Member Broached or Shaped

A 30-degree involute broach or shaper cutter is used to process internal splines for all types and classes of fits as well as for parallel-root and taper-root splines. A different broach or shaper cutter is required for fillet root type and flat root type. The shaper cutter can be used for cutting any number of teeth of the same diametral pitch while a broach will cut only the number of teeth for which it was designed. Broaches are usually used only on large production jobs. The broach or cutter has a ramp or minor diameter chamfer inside the true involute form which produces the required clearance for the fillet on the external member.

STRENGTH ADVANTAGES OF TAPER-ROOT DESIGN: The 30-degree involute spline has inherent strength advantages over the parallel-key design. The tooth has maximum strength through the minor diameter where it is actually needed most. The minor diameter of an involute spline is usually greater than that of a parallel-key spline of the same outside diameter. This is because the stub-tooth form is considerably shorter than the parallel-key form. Under the new standard,* it is possible to increase the number of teeth while maintaining the original major diameter. This increases the torsional capacity of the shaft because the minor diameter increases as the number of teeth increases.

Because the depth of the taper-root spline is reduced to zero at the large end of the taper, the torsional strength of the shaft is greater than for the parallel-root design. In fact, the strength of a taper-root involute splined shaft is approximately equal to that of a solid shaft with a diameter equal to the pitch diameter of the spline.

DESIGN OF TAPER-ROOT SPLINES: The strength of the spline teeth is not always a vital factor to consider in the design of a taper spline. If a taper-root involute spline is to replace a taper-root parallel-key spline, the same length of involute spline can be used because it is stronger than the parallel-key design. In many cases, the design has dictated a shaft much larger than the maximum load (with an adequate safety factor) would require. Here, a taper-root involute spline of the same length as a parallel-root involute spline should be satisfactory. Also, it should be remembered that the torsional capacity of the

* *SAE Handbook*, 1950.

shaft is greater with the taper-root involute spline than with any other spline.

However, there are many new applications where the spline must carry the full torsional capacity of the shaft. To furnish a basis for determining the length of the spline which will handle this load, a strength formula for the taper-root involute spline has been developed.

Determining Spline Length: The derivation of the formula for figuring the length of a parallel-root involute spline can be found in any recent edition of the *SAE Handbook.* Based upon the bearing compressive torsional strength capacity of the shaft, bearing stress and torsional shear stress, the formula which applies to most involute splined shafts is (see Nomenclature)

$$L = D_r^3/D^2$$

Fig. 3 illustrates the basic assumptions from which the formula is derived for figuring the length of key that will apply to most taper-root involute splined shafts. As can be seen in *Fig.* 3, the height of contact on the tooth is

$$0.9h = 0.9D/N$$

The total height of the tooth having been established equal to D/N, the reciprocal of the diametral pitch. The area in compressive stress for one tooth becomes, from *Fig.* 3,

$$A_c = \tfrac{1}{2}\,(0.9h)\,(0.9L) = 0.405DL/N$$

or for purposes of simplication

$$A_c = 2DL/5N \text{ (approximately)}$$

The torque which a taper spline will deliver

Nomenclature

Q = Torsional moment, lb-in.
Q_c = Torque based on compressive stress, lb-in.
Q_s = Torque based on shear stress, lb-in.
S_s = Allowable torsional shear stress, psi
S_c = Allowable compressive bearing stress, psi
D = Pitch diameter of spline, in.
D_r = Minor diameter of spline, in.
A_c = Area in compressive stress for one tooth, sq in.
A_s = Area in shear stress for one tooth, sq in.
T = Maximum width of inclined section at pitch line, in.
N = Number of teeth
L = Length of spline, in.
h = Height of tooth, in.

based on compressive tooth stress, with each tooth carrying its share of the load, is

$$Q_c = \frac{S_c ND(2DL/5N)}{2} = \frac{S_c D^2 L}{5} \qquad (1)$$

The area in shear at the pitch line in a parallel-root spline is $\pi DL/2$. In an involute taper spline, the area in shear is A_s as shown in *Fig.* 4.

$$A_s = 0.5TL + \tfrac{1}{2}\,(T + 0.6T)\,(0.5L) = 0.9TL$$

With an allowance for the chamfer and relief at the top and bottom of the tooth, the effective length will be reduced to $0.8L$, *Fig.* 3. Therefore, the actual area in shear in one tooth is

$$A_s = 0.8(0.9TL) = 0.72TL$$

The total area in shear in a complete spline having

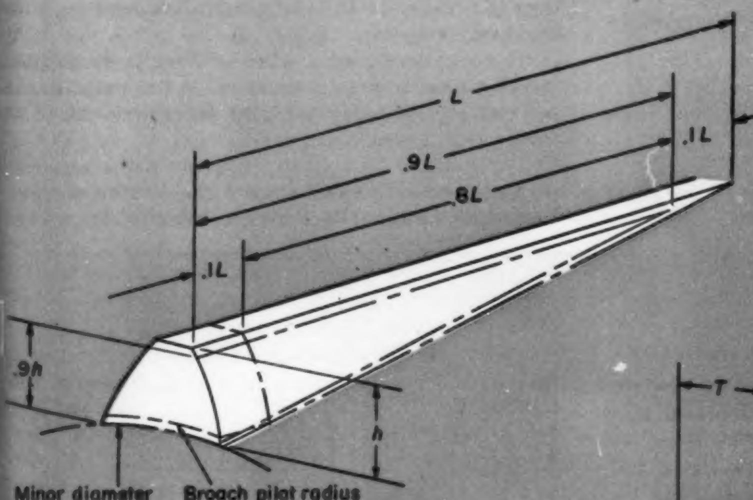


Fig. 3—The compressive tooth stress is critical on a taper-root involute spline. One-tenth of the height of the spline at the outer end of the taper is considered as having no contact because of the chamfer at the top and root of the spline tooth. Definitions of the symbols are given in the nomenclature

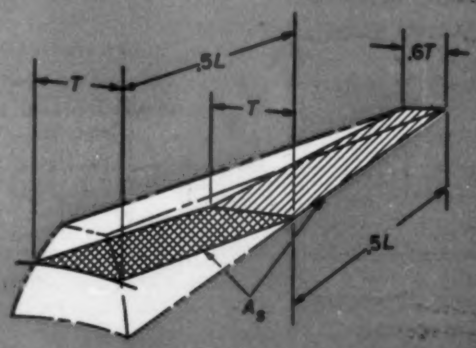


Fig. 4—The area in shear stress is indicated by the shaded lines. It will be noted that the width of the tooth, T, decreases from full width, where the pitch line contacts the root, to 0.6T at the inner end of the taper

N teeth is therefore $0.72NTL$, assuming perfect contact. Since $NTL = \pi DL/2$,

$$A_s N = \frac{0.72\pi DL}{2} = 0.36\pi DL$$

The torque in shear on a tapered spline becomes

$$Q_s = 0.36\pi DLS_s D/2 = 0.18\pi D^2 S_s L \quad (2)$$

Equating the torques in compression and shear, Equations 1 and 2,

$$0.18\pi D^2 S_s L = \frac{S_c D^2 L}{5}$$

from which the ratio of actual stress is

$$\frac{S_s}{S_c} = \frac{D^2 L}{5(0.18\pi D^2 L)} = 0.354$$

If the allowable shear stress is 60 per cent of the tensile ultimate strength, and the allowable compressive stress is 140 per cent, the desired ratio is

$$S_s/S_c = 60/140 = 0.428$$

Dividing this ratio by the ratio of actual stresses,

$$0.428/0.354 = 1.21$$

which is the allowable/actual ratio. It can be seen that for a specified allowable shear stress, the accompanying compression stress exceeds that permissible. Hence the compressive stress, rather than the shear stress, is critical.

Since a parallel-root involute spline is stronger than a plain shaft the diameter of which is equal to the root diameter of the splined shaft, it would be conservative to evaluate the torque capacity of a tapered spline shaft as

$$Q = \pi D_r^3 S_s/16 \quad (3)$$

Setting Equations 2 and 3 equal and solving for L.

$$0.18\pi D^2 S_s L = \pi D_r^3 S_s/16$$

$$L = D_r^3/2.88D^2$$

This length is based upon shear stress, but since the compressive stress is the critical stress, L must be multiplied by 1.21, the allowable/actual stress ratio. The corrected length based on the critical compressive stress becomes.

$$L = 1.21D_r^3/2.88D^2 = 0.420\ D_r^3/D^2 \quad (4)$$

This equation for determining the length of spline applies *only* under ideal conditions. Since perfect parts cannot be made and since spacing errors will cause the load to be concentrated on a few teeth at first, it is well to allow for such eventualities.

Using the same safety factor of 4 which is used in the *SAE Handbook* for parallel-root involute splines, the length becomes

$$L = 1.680\ D_r^3/D^2 \quad (5)$$

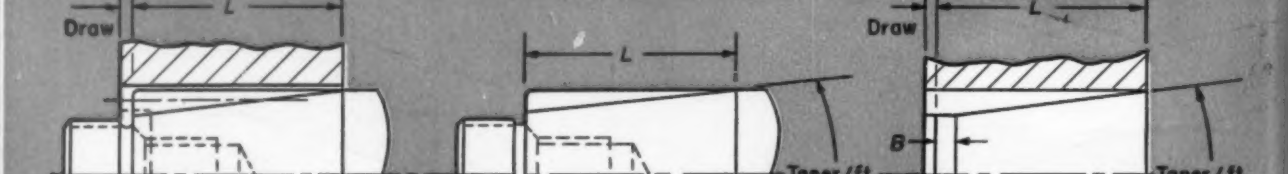
This safety factor should be large enough to compensate for errors in spacing, other inaccuracies in machining, type of material, surface finish, etc. Thus, it appears that an involute taper spline should be longer than a similar parallel-root involute spline if it is to carry the full torsional capacity of the shaft.

This does not necessarily mean that the taper spline must be 1.680 times as long as the parallel-root involute spline. The numerical factor remains constant, but $D_r^3D^2$ is a variable which decreases as the diametral pitch becomes coarser. Ordinarily, the outside diameter of the shaft and the amount of taper per foot for the root of the spline are established by the design and the application. However, the diametral pitch can be varied to accommodate these fixed elements.

Determining Taper and Other Details: The taper-root involute spline requires certain elements of design which are not encountered with parallel-root splines. To be considered in addition to the length of the splined shaft are factors such as amount of taper, ratio of depth to length, standard dimensions to be used, etc.

These splines are engineered from a basic plane located from a gaging shoulder on the part. A land or flat should be provided at the small end of the

Fig. 5—The length of the hub is equal to the length of spline plus the draw. Amount of draw is a function of type of material, hardness of the material, rate of taper and amount of pressure used to draw the hub onto the spline



tapered hole to serve as a guide for the pilot of the broach. All processing of the internal member is gaged from this land.

There is no rule for determining the amount of taper for the root of the spline. The normal rate is $1\frac{1}{2}$ inches per foot (included angle). This provides a good base for an assembly which does *not* have to be easily removed and normally provides enough length so that the spline keys will be strong enough to carry the full torsional capacity of the shaft. It might be called a semipermanent fit since the hub can be removed, but not too easily. It is also satisfactory for drawing the hub onto the spline to take up any wear.

Taper To Suit Design

As the amount of taper is increased, it becomes easier to remove the hub. For parts which must be easily removed, a taper of 3 inches per foot (included angle) is often recommended. It must be considered, however, that this amount of taper will seldom allow length enough so that the spline will carry the full torsional capacity of the shaft. In this case, the shaft will have to be considerably larger than one required to carry the maximum load times the safety factor. It is often possible to change the amount of taper slightly to suit other desired design requirements which cannot be met in any other way. Typical included tapers used range from $1\frac{1}{2}$ to 4 inches per foot in $\frac{1}{4}$-inch increments.

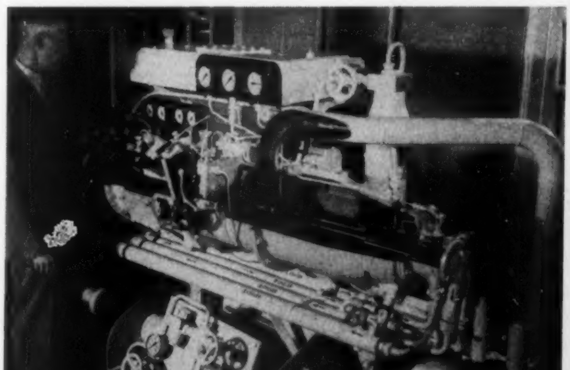
Since the length of cut also depends upon the depth of cut, spline length is approximately inversely proportional to the pitch. For example, the length of cut for an 8/16-pitch spline would be approximately one-half that for a 4/8-pitch spline.

The amount of draw is the distance which the spline will move from a light fit until it is drawn up tightly on the hub. The purpose is to prevent the spline from extending beyond the hub. The amount depends upon the type of material, the hardness of the material, the rate of taper and the amount of pressure which is used to draw the spline onto the tapered root of the hub. If wear is to be taken up on the taper, the amount of wear will have to be estimated so that it can be added to the amount of draw. If the actual amount of draw should exceed the amount anticipated in the design, a washer could be inserted between the hub and the nut so that wear could still be taken up on the taper.

Either the flat root or full fillet type tooth may be used in the internal member of the taper-root spline. The external member, however, must be confined to the flat root application in order to produce as great a root bearing as possible as well as to minimize the engineering procedure.

Tests Prove Compressor Efficiency

UNUSUALLY high efficiency is one of the characteristics of the Junkers free-piston diesel air compressor used aboard German destroyers and submarines during the war as a source of high-pressure air and shown in the accompanying photograph. Tests at the U. S. Naval Engineering Experiment Station at Annapolis, Md., have shown the unit to be reliable and well suited to shipboard installations. The free-piston compressor differs radically from the conventional type in that there is no crankshaft or flywheel. Reciprocating motion is obtained from the energy of combustion in one direction and from the potential energy of the compressed air in the other direction. Power is obtained from a horizontally-arranged, opposed-piston, two-cycle, uniflow diesel engine. The two pistons are synchronized by means of two sets of racks which operate on two fixed pinions located at the center of the machine. The pinions act as drives for the accessory pumps.



Free-Piston Versus Conventional Compressors

	Junkers	Mfg. A	Mfg. B	Mfg. C	Mfg. D
Compressor strokes/min..	877	550	545	300	300
Total weight, lb	1210	4120	5176	6447	8547
Working pressure, psi...	2700	3000	3000	2500	3000
Capacity, cfm @ 80 F and 14.7 psia	55.93	69.04	68.03	81.48	75.77
Weight, lb/cfm of air...	21.6	59.7	76.1	79.1	112.8
Fuel consumption, lb/hr.	17.36	22.49	22.69	20.54	22.01
Cu ft air/lb of fuel	193.3	184.2	179.9	238.0	206.6

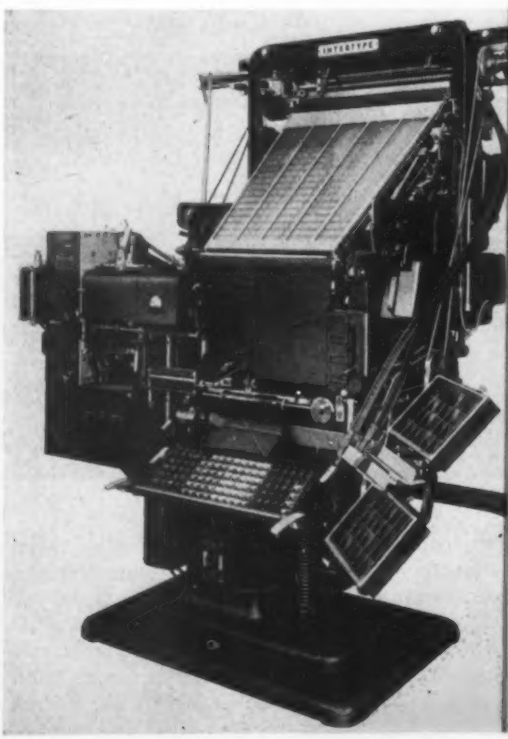
The accompanying comparison of the Junkers unit with various other compressors of similar capacity and discharge pressure shows it to be light and efficient and to have a high output for its weight and volume.

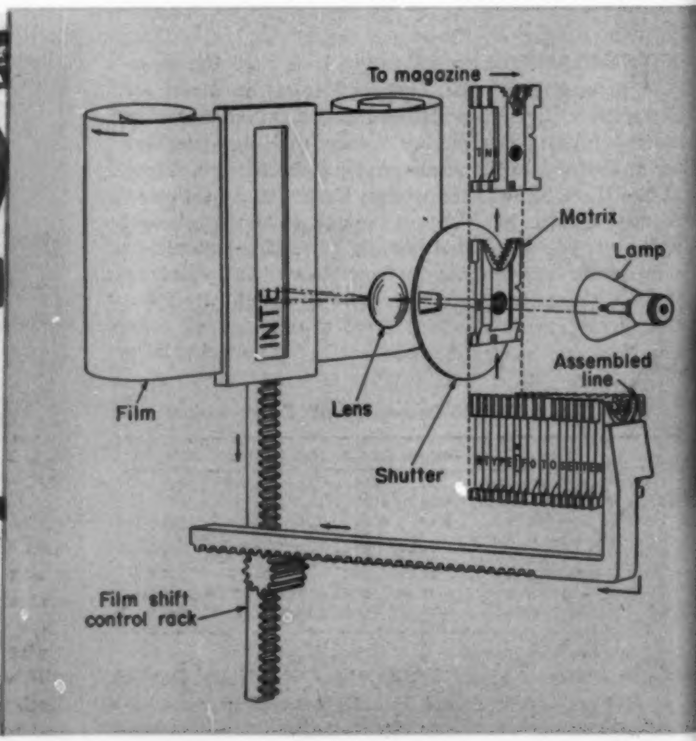
Contemporary D

OUTSTANDING strides in the design of machines and equipment for the printing industry are noteworthy, especially with respect to refinements making possible higher production speeds and better quality work. Reciprocating mechanisms have in some cases yielded to continuous motion. Shapes and forms have been styled pleasingly without sacrificing accessibility. New machines developed for offset printing make the process more direct, more economical and faster while improving the quality of the product. This review illustrates the trends evident in new equipment for the printing industry as shown at the recent Graphic Arts Exposition in Chicago.

Composing Machine Employs Photographic Principle

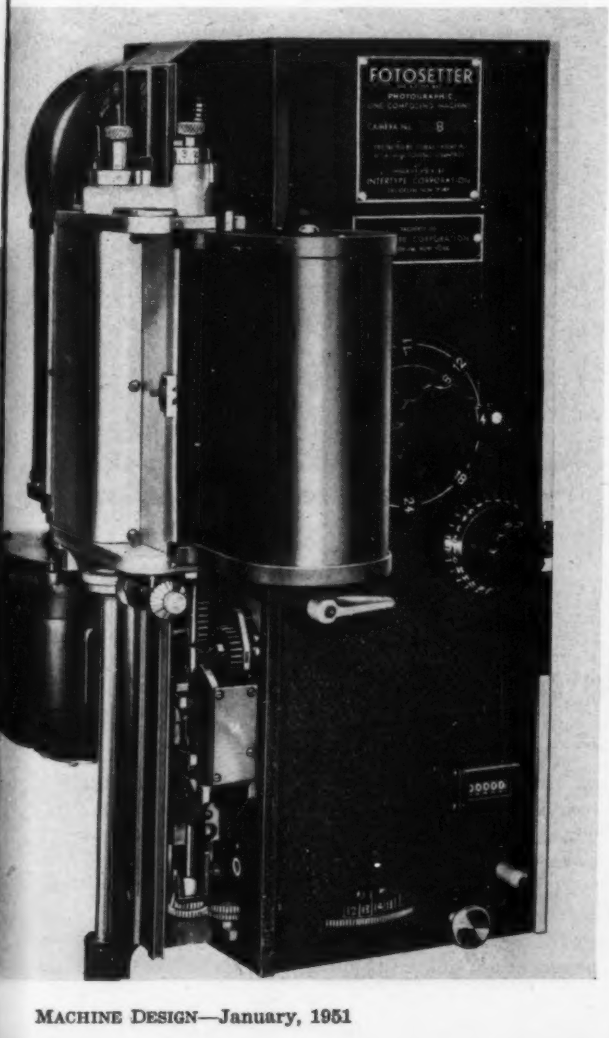
PHOTOGRAPHIC line composing machine, below, produces justified composition directly on film or photographic paper so that plates for offset gravure or letterpress printing can be made from the photographic composition using standard platemaking methods. This machine obviates the usual processes of type casting and proofing, eliminating the usual losses through these intermediary processes before making a photographic copy. Made by Intertype Corp., this Fotosetter uses the circulating matrix





SIGN

principle of linecasting machines but replaces the metal pot with a camera. The Fotomat or matrix, instead of carrying a character mold in its edge, has a transparent character imbedded in its face, as in the drawing, previous page. As in standard linecasting machines, matrices are assembled in any line length under 42 picas, and lines measured for justification. The camera then photographs the projected images of the matrices one at a time, with proper justifying space automatically distributed between words and characters as desired. The film carriage, photograph below, drops by gravity each time a matrix is fed into the camera. Proper spacing is

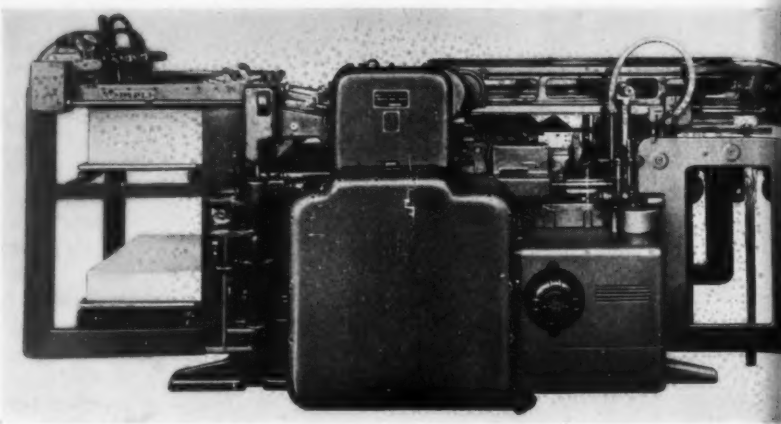


obtained by racks and a gear train controlled by the thickness of each character as shown in the drawing, and by a cam mechanism which compensates for short lines. Each matrix is held stationary before the lens while the exposure is made to provide a clear image. After a line has been completely photographed, the film advances automatically to give any preset line spacing up to 36 points. Eight lenses mounted in a rotating turret, combined with the four magazines, give 32 different type face sizes without changing magazines.

Machine Rules Lines on Forms

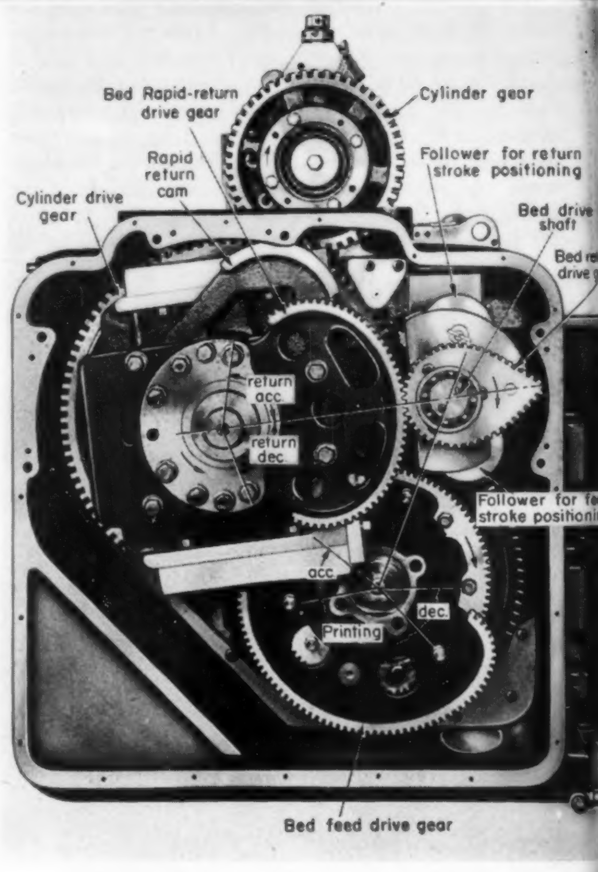
A NEW machine, below, rules fine lines and leader lines for form work and sets justified copy for these forms in the same operation. Made by Ralph C. Coxhead Corp., the DSJ Forms Design machine rules single or double lines in dots or dashes. The method of inscribing a line is the same as imprinting any character on the font. The type of line to be inscribed is selected by depressing its key and the impression button, the character automatically repeating until the line is formed. Length of the line is predetermined by stops. Width of the spaces between lines can be varied and adjusted to any desired spacing chosen according to the printer's point-leading system. When the carriage is returned for the next line-ruling the uniform width-spacing is automatically set.

Press Bed Driven by Eccentric Gearing



ROLLCURVE gearing, an innovation in press drives, reciprocates the bed in the No. 29 Letterpress, above, made by Miehle Printing Press & Manufacturing Co. All gear trains in the drive operate at the same angular speed with the exception of the bed drive gear assembly, which oscillates. Principle of the unusual drive is the use of rollcurve gears and cams shown in the illustration, right. The gear train is shown in the rapid-return part of the cycle. At the beginning of the return stroke, the return cams have contacted cam followers on the bed-drive gear shaft to position the eccentric drive gear with the heart-shaped gear which is being driven in a clockwise direction. The illustration shows the drive at the midpoint of the rapid-return stroke. The return rollcurve gearing provides for equal periods of acceleration and deceleration with no unnecessary constant-speed period.

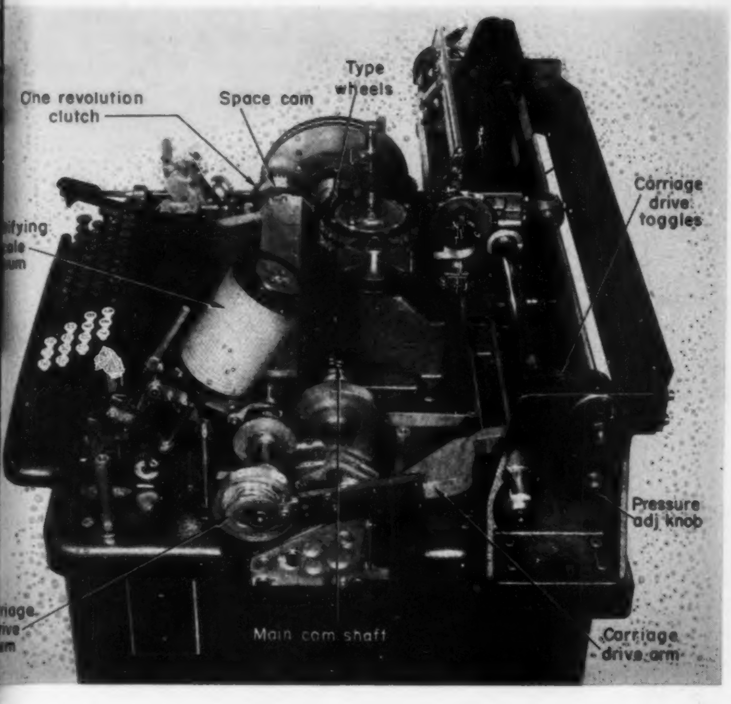
After the return stroke is completed, the cam followers on the bed drive gear shaft contact cams (partly shown) on the cylinder drive gear shaft assembly to engage the gears for the print stroke. This gearing drives the bed in the printing direction, providing a short acceleration period, then a longer constant-speed printing period ending with a short deceleration. Exact rolling contact is maintained between the impression cylinder and bed during this printing period because both are driven from the main drive shaft. The press bed is driven from the bed drive shaft through a conventional rack-and-gear arrangement. This press handles a maximum sheet size of 22 by 28 inches at 4500 impressions per hour.



Type Variety Increased in Justifying Composer

THE Lithotype Composer manufactured by the Fairchild Camera and Instrument Corp., top of next page, combines extreme flexibility in the selection of conventional printing type faces with the economy of standard typewriter operation. It provides type composition for all forms of photo-offset reproduction and also may be used to advantage in certain types of letterpress work. Type is mounted on magnesium-alloy wheels in two rows permitting combinations of different type faces. These type wheels may be readily changed and damaged or worn characters may be quickly replaced.

The impression on the work sheet is made by the carriage being moved forward to the typewheel. The work sheet—which may be white-coated paper, a paper offset plate or an aluminum offset plate—is

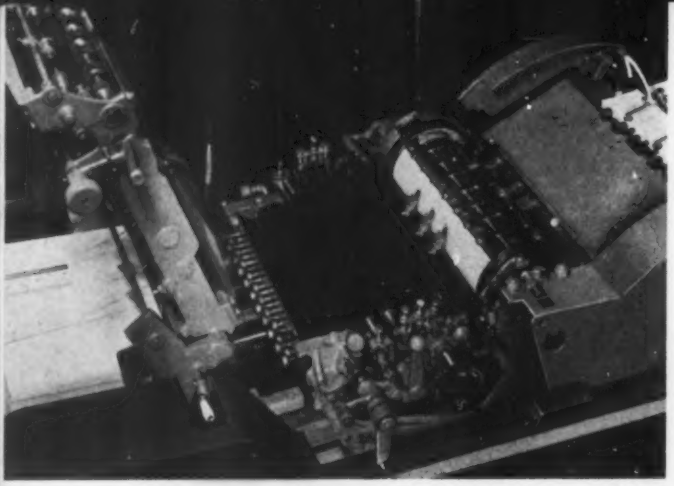


backed by a flat nylon bar in the printing area and, since the type is also flat, the impression is uniform from top to bottom of each character. The drive arm which moves the carriage to and from the typewheel or print position, view with cover removed, above, is actuated by a cam fixed to a one-revolution shaft. The amount of travel is governed by the throw of the cam; however, pressure adjustments between the type wheel and carriage are made by rotating the pressure knob to one of ten positions. This knob rotates an eccentric in the end of the drive arms which, in effect, lengthens or shortens the drive arm by the amount of eccentricity. Position of the carriage-drive toggles also changes when the adjustment is made and the print position of the carriage or impression changes automatically.

Timing sequence for the various operations after a key is depressed are fixed with relation to each other. When a key is depressed, a stop lever engages the particular cam on the main camshaft, stopping the typewheel, which is geared to the camshaft, in approximate position for the character. The stop lever also trips the one-revolution clutch. As the shaft rotates, the register bolt engages a register wheel at the lower end of the typewheel shaft and locates the character on the typewheel in the exact printing position. Then the carriage drive cam moves the carriage forward to make an impresion and returns. The space cam then releases an arm carrying pawls which pick up the correct number of teeth or units on the space ratchet wheel for the character printed.

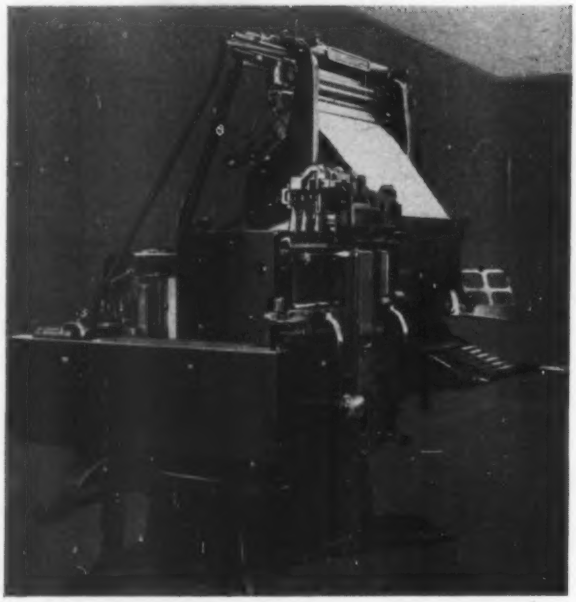
Makeready Simplified in Cylinder Press

ACCESSIBILITY of type on the press bed, high speed and modern styling are features of the Chandler & Price Co. cylinder press illustrated at the left. With a speed of 2200 to 4800 impressions per hour in sheet sizes from $12\frac{1}{4}$ by $18\frac{1}{2}$ inches down to $3\frac{1}{4}$ by 5 inches—onion skin to 4-ply cardboard—the press is adaptable to long or short-run work. Sheets are vacuum-registered accurately with a mi-



cromometer side adjustment before being transferred to the cylinder. The cyinder is completely accessible for makeready without removing any press parts and, when the press is open, the type form is exposed and can be unlocked and relocked for making changes directly on the press bed. Photograph with the hinged roller carriage lifted, left, shows the 10 distribution rollers from fountain to form, including two form rollers, three vibrators and two distributors. Simplified operating controls are grouped on the working side of the press for convenience of the operator.

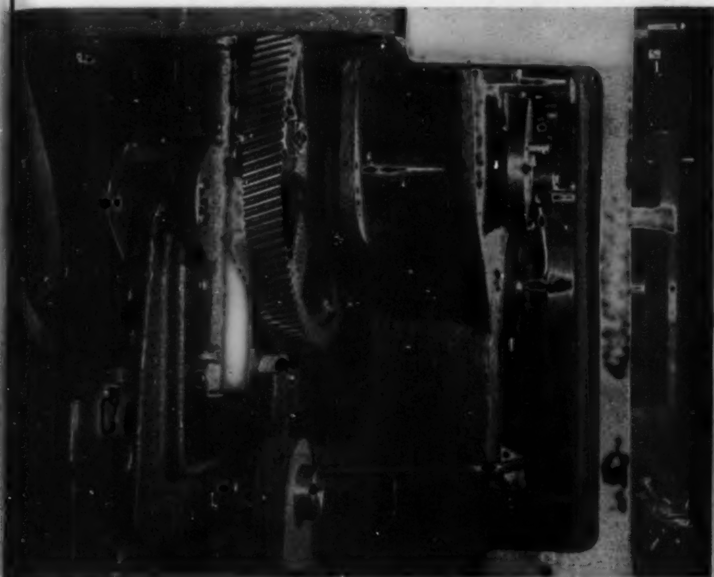
Photo-Composer Simplifies Offset Printing

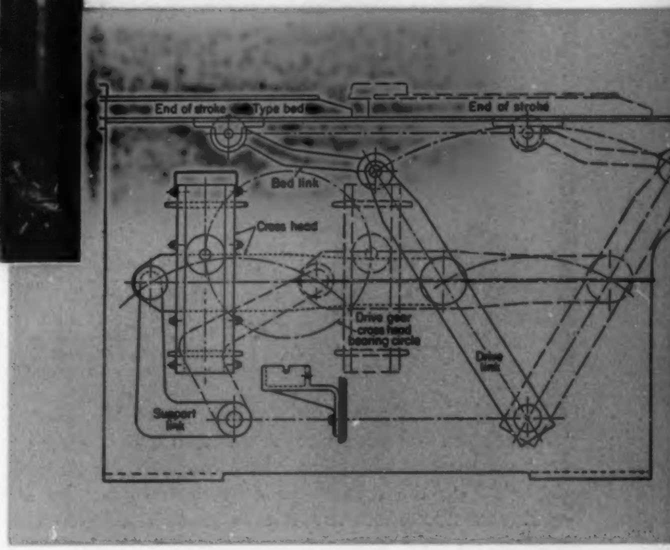


OPERATING on a modified Linotype principle, the two-magazine Linofilm photo-composing machine, left, produces up to eight lines of filmed copy per minute. The machine is similar to the conventional linecasting machines insofar as distribution, storage, shape of matrices and line justification are concerned. However, the usual molding equipment has been replaced by a camera that photographs an entire line at a time. Either film or paper with black characters on a clear background are produced for making plates or gravure cylinders. Operating on 12-point, two-letter matrices, the machine reduces or magnifies this type to give 12 sizes from 6 to 36-point. Spacing between lines is variable between 1 and 12 points. Developed by Mergenthaler Linotype Co., the machine also includes a stop to prevent the last line photographed from being redistributed until composition is resumed.

Press Drive Permits Higher Speeds

VERSATILITY is the keynote of the new Model 6 Little Giant press, left, which handles any stock from onion skin to 0.018-inch, 4-ply at speeds between 2500 and 5000 impressions per hour. Eight feeder suckers and three blast pipes give improved sheet pickup and a new form roller arrangement, including three form rolls, three vibrators and two distributor

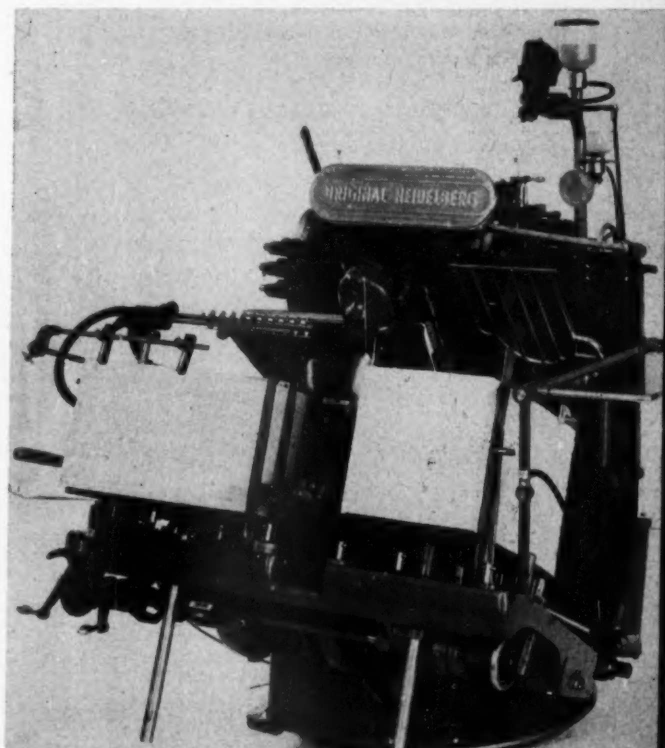


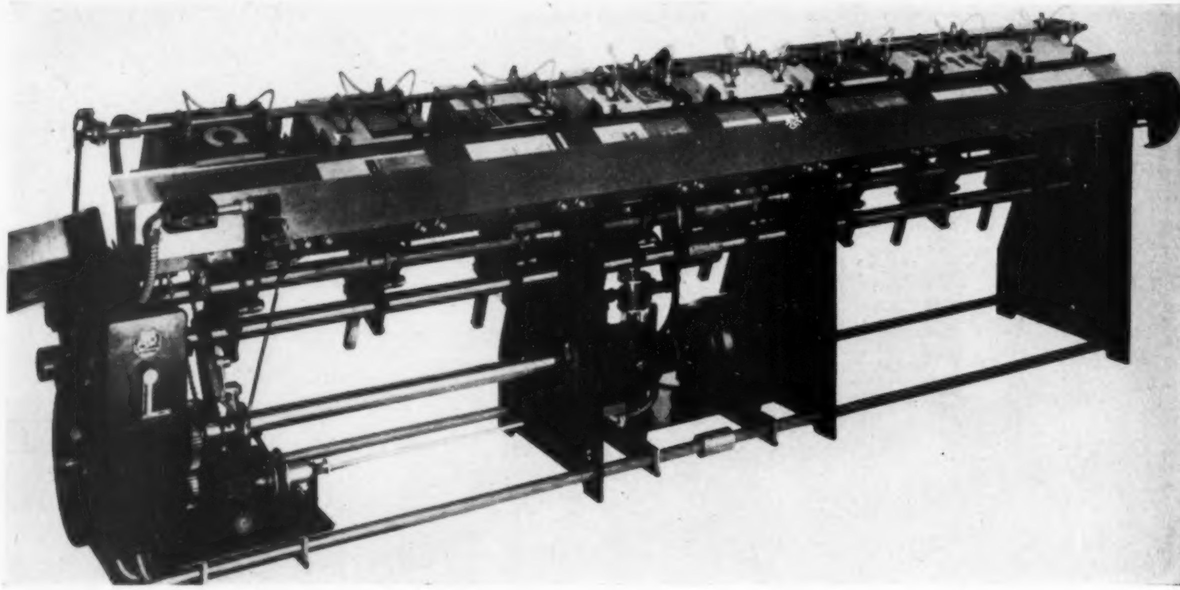


rollers, gives better ink distribution. A new cylinder and bed drive incorporates a slide ball bearing instead of a direct connection between crank arm and the driving gear. The photograph, above, and drawing, right, show details of this drive which permits higher speeds and a smoother motion than was previously possible. Made by American Type Founders, the press includes stepless speed variation obtained from a variable-speed pulley, belted to a 2-hp motor.

Platen Press Has Rotating Feeder

AN UNUSUAL two-arm, propeller type sheet feeder is one of the interesting features of the 12 by 18 Original Heidelberg automatic platen press shown at the right. The feeder arm mechanism is attached to the pivoted platen, with the arms rotating in a plane parallel to the plane of the platen. Sheets are separated from the stock pile by air blasts from three sides, then grippers on the feeder arms pick up the sheets and pull them into the open press. After the impression has been made, the same arm picks up the printed sheet and pivots, pulling the sheet to the delivery pile. An outstanding feature of this press is the built-in roller washer which automatically cleans all the rolls in one minute. A lever disconnects all rollers simultaneously, eliminating time formerly required to take out individual rolls when the press is down. Any speed between 1300 and 4000 impressions per hour is obtainable and a toggle lever arrangement gives adjustable impression pressures up to 60 tons for die-cutting or embossing.



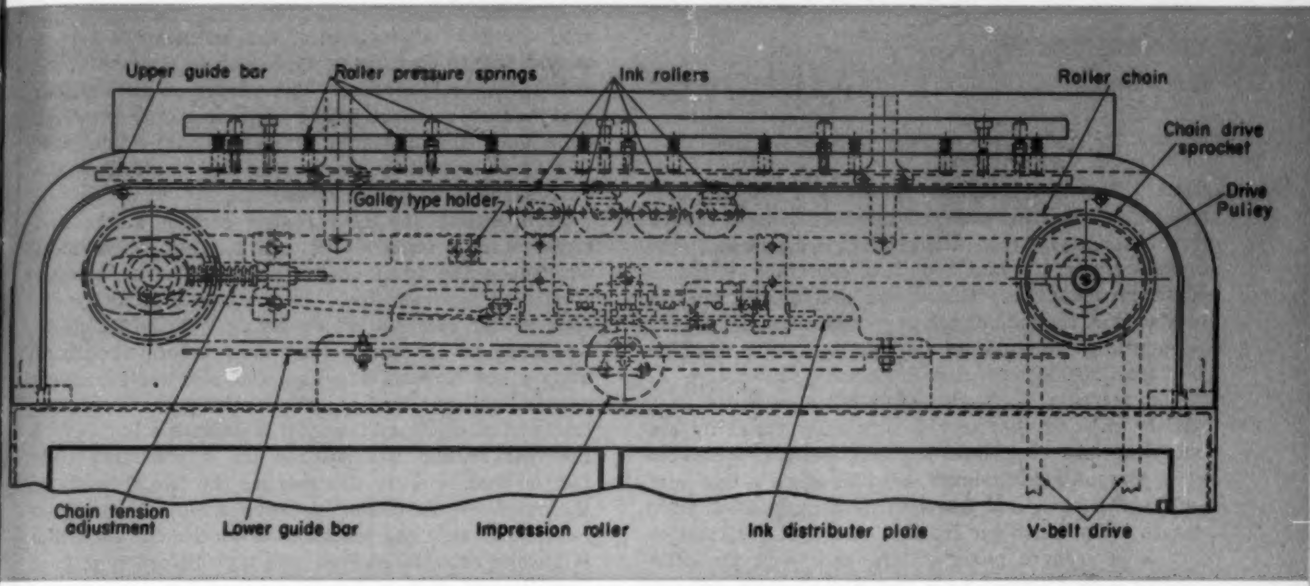


Detector Checks Collator Gathering

SIMPLICITY is the design feature of the Macey collator, above, capable of gathering 25,000 sheets an hour into sets of eight. Manufactured by Harris-Seybold Co., the collator uses suction to pick up, simultaneously separating sheets from others in the pile with a blast of compressed air. As an additional precaution, an automatic detector checks each completed set, stopping the machine if there is an extra sheet or if one is missing. Built on the conveyor principle, the collator consists of eight hoppers or stations facing a slanted stainless steel tray. Each station will accommodate a stack of paper 11½ inches high, about 3000 sheets of 20-pound stock. Although the machine will collate up to eight sheets per set, individual stations can be turned off and any number of sheets from two to eight can be gathered. Maximum sheet size is 9½ x 12 inches with a minimum of 5 x 8½ inches. The machine handles stock ranging from 9 pound to ⅛-inch cardboard, both plain and punched. Stations are individually self-leveling, keeping the top sheet at proper height for separating and feeding regardless of weight of stock. Drive to the sheet feed mechanism is through a pivoted rocker arm driven by a cam and closed cam track in the flywheel. A separate motor drives an air pump for sheet separation and suction pickup.

Proof Press Uses Continuous Motion

CONTINUOUS one-way movement of impression and ink rollers in the proof press, left, replaces the usual reciprocating motion of impression cylinder and rollers. Four ink rolls and one impresion roller are carried by continuous roller chain driven through variable-pitch sheaves at from 25 to 50 impressions per minute. Rollers are held in contact with the type at

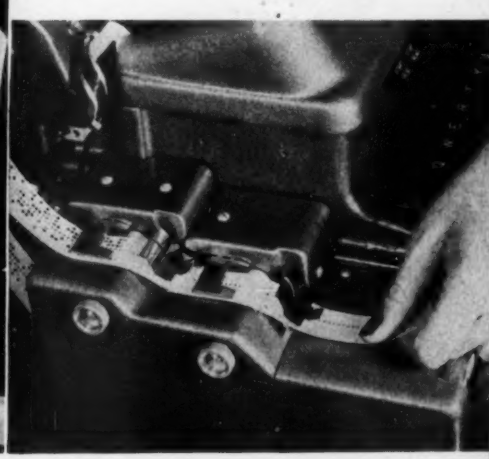


the correct pressure by spring-loaded upper guide bars, drawing above. Lower guide bars, underneath the bed of the press, hold the impression roller out of contact with the rotating ink distributer plate while guiding the ink rollers into contact with the plate. Controls stop the press after each cycle to prevent overrun of the impression and ink rolls. If desired, the automatic stop controls can be bypassed to give continuous operation of this press which is built by Wesel Manufacturing Co.

Contemporary DESIGN

Reader Unit Automatically Justifies Copy

A COMPOSING machine that automatically produces justified lines with one manual keyboarding is illustrated at the left. Developed by Commercial Controls Corp., the Justowriter consists of two companion units, a recorder and a repro-



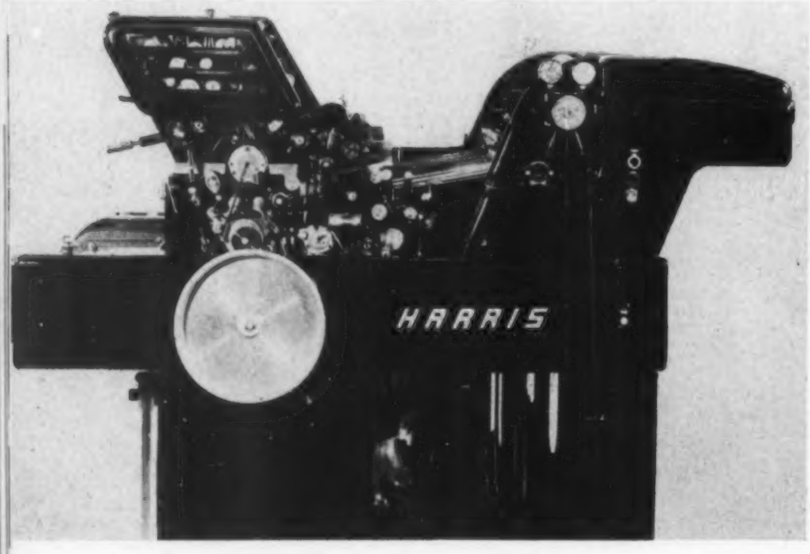
ducer. The combination is basically a proportional-spacing, electrically-powered machine with standard typewriter keyboards. The product is a justified page or galley in a 12 point type face especially suited to offset reproduction. Characters and spaces are reproduced at the rate of ten or more a second. Actual automatic production on the justified copy machine is at least 20,000 characters per hour.

The recorder automatically perforates codes in a narrow tape as the copy is originally typed by the operator. As each line is typed, a light controlled by a computer mechanism indicates when a line is of sufficient length and has sufficient number of word spaces to permit justification. Then a justification key is pressed to perforate the tape with an interlocking code followed by a justification code selected by the computer mechanism. When the tape is traveling through the reproducer the justification code is sensed, before starting to type each line, by the rear unit in the double reader. This reader consists of two independent sections, photo bottom right, previous page, operating at spaced points along the same tape. The front section is for reading and controlling the automatic printing mechanism.

In operation the rear reader section starts and reads until it finds and stores a justification code, whereupon the front reader section is automatically started to type the line. As the line is being typed, both reader sections are operating. If the rear section reaches an interlocking code before the front section has finished the line, the rear section stops automatically at that point. Similarly the front section will stop automatically if it finishes a line before the rear section has reached the interlocking code for the next line. In this manner the two sections of the double reader of the Justowriter reproducer are so interlocked that the justification control for each line is always established from the tape before typing of that line is started regardles of variations in relative lengths of lines.

Offset Press Features Ease of Adjustment

DESIGN advances in the Harris-Seybold Co. 17 by 22-inch offset press shown below increase accuracy of register and operating convenience at speeds to 7000 sheets per hour. Stream or single sheet-by-sheet feed is optional, with automatic slow-down at the front guides. The press is driven through the delivery cylinder to provide an even flow of power to the printing and inking mechanisms. To allow free access to the registering mechanism and to the dampeners, the fountain slides back seven inches from the printing position. The photograph of the back of the press with the guards removed, below, shows the 20 rollers between fountain and plate that assure smooth lay of tacky inks. A pinion and internal gear are used to rotate the plate cylinder (seen behind the chain at the right center of the closeup photograph) independently of the drive gear to change or adjust the front lay. Four form rollers are accessible for adjustments from outside the press, facilitating the makeready operation.



Designing for SHOCK RESISTANCE

Part 2

By
Charles E. Crede
and
Miguel C. Junger
The Barry Corp
Cambridge, Mass.

SHOCK resistance concepts require not only that equipment remain free from damage that prevents it from performing its intended function, but also that the equipment operate properly during and after the shock. Mechanisms must necessarily be designed so as not to be accidentally operated as a result of shock. This second of a two-part series sets forth several basic principles which contribute toward attaining this condition.

BALANCED MECHANISMS—ONE PIVOT: When machinery is subjected to an acceleration, the movable members tend to remain stationary in space because of their inertia. The chassis moves in response to the applied acceleration, and relative motion tends to develop between the chassis and the movable members, causing maloperation of the equipment.

Information for these articles has been provided by the Bureau of Ships, Department of Navy.

In mechanisms embodying one pivot, static balancing of the mechanism with respect to this pivot will prevent maloperation from accelerations having only translatory components. A mechanism in static balance has equal products of mass and moment arm on opposite sides of the pivot. When the equipment is subjected to translatory acceleration, the resultant of the couples created by the inertia forces is then equal to zero, and there is no tendency for accidental movement of the mechanism relative to the chassis to take place.

An example of a statically balanced mechanism with a single pivot is a circular type rheostat, shown schematically in *Fig.* 15. The resistance element is wound upon an annular form with the moving contact carried on the end of an arm pivoted at the geometric center of the resistance element. A counterbalance is provided to balance the arm statically with

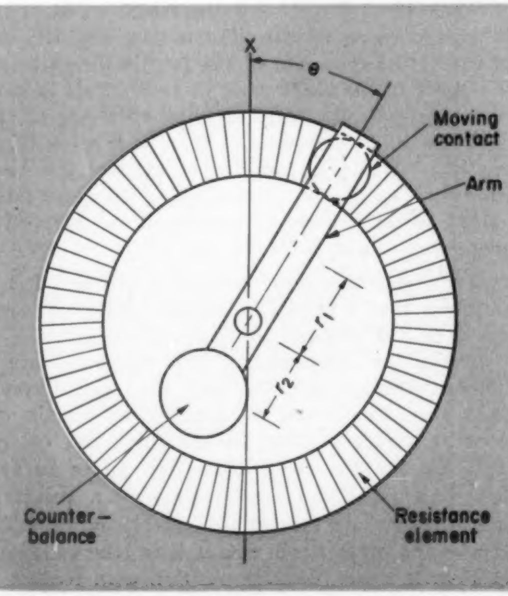
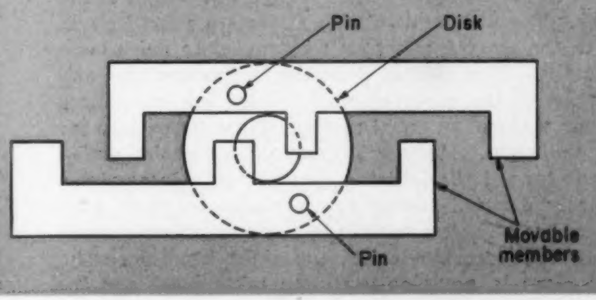


Fig. 15—Left—Circular type rheostat with statically-balanced single-pivot mechanism

Fig. 16—Below—Static balance mechanism using identical members moving in opposite directions



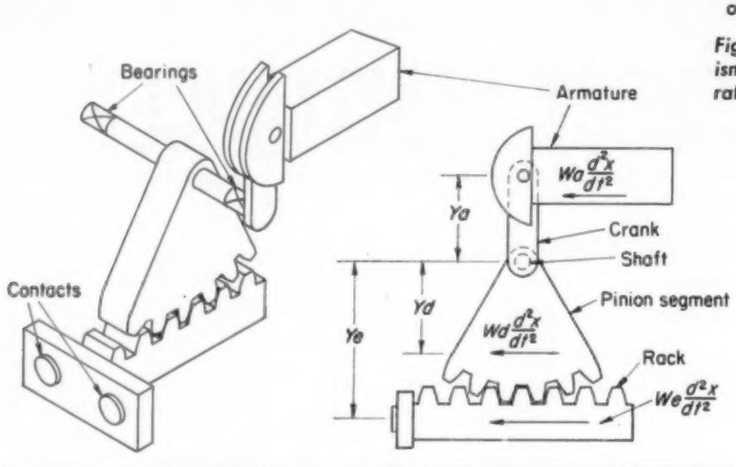
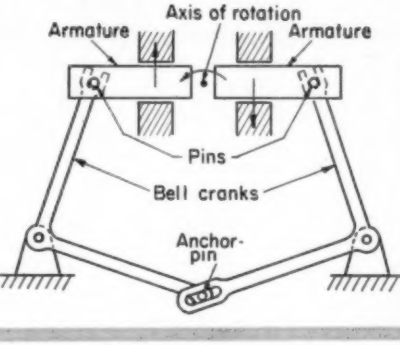


Fig. 17—Left—Perspective and schematic views of statically-balanced contactor mechanism

Fig. 18—Below—Multiple-pivot balance mechanism for balancing against rotational acceleration. Inertia forces and couples are balanced around axis of rotation



respect to its pivot. The arm has weight W_1, whose center of gravity is at radius r_1 with respect to the pivot. The counterweight W_2 is located at a radius r_2. If the rheostat is now subjected to a translatory acceleration d^2x/dt^2 in the x direction, which is at any angle θ with the direction of the arm, the resultant inertia couple M acting on the arm may be written

$$M = \frac{W_2}{g}\frac{d^2x}{dt^2}r_2\sin\theta - \frac{W_1}{g}\frac{d^2x}{dt^2}r_1\sin\theta \qquad (1)$$

If the position of the arm is not to change, it is necessary that the couple M be zero at all times. Dividing both sides of Equation 1 by $(d^2x/dt^2)(\sin\theta)/g$, and setting M equal to zero,

$$W_2r_2 = W_1r_1 \qquad (2)$$

The criterion given by Equation 2 is that for the static balance of a single pivot mechanism. It may be extended to an arm comprised of any number of discrete weights. The physical significance of the criterion is that the axis of rotation must pass through the center of gravity of the arm. It should be noted that Equation 2 is independent of the angle θ, and therefore applies to all accelerations comprised only of translatory components.

This same principle is illustrated by the mechanism shown in *Fig.* 16, which includes two identical horizontally movable members with a pin attached to each of the members and extending into a well in the disk. The disk rotates with respect to a fixed pivot, with the moving members constrained to move horizontally. It is evident from the mechanism that any motion of the upper member toward the left cannot occur without an equal motion of the lower member to the right. An acceleration of the entire equipment toward the left, for example, introduces rightwardly directed inertia forces associated with both members. The constraints of the mechanism prevent both members moving to the right simultaneously. As a result, no relative motion takes place and the mechanism becomes unresponsive to translatory accelerations.

Static balance principles also find application in the contactor illustrated schematically by the perspective view of *Fig.* 17. The contactor is operated by energizing a solenoid and the resulting magnetic force induces a motion of the armature, which is changed by a shaft and crank to rotational motion of the pinion segment. The pinion engages a rack which carries the electrical contacts, the rack being moved generally parallel with the armature but always in the reverse direction.

Inertia forces which influence the response of the contactor to shock are shown in *Fig.* 17, in the view in a plane perpendicular to the shaft. Assume the contactor to experience a rightwardly directed translatory acceleration of magnitude d^2x/dt^2. The inertia forces associated with the various massive elements, all directed toward the left, are indicated by the arrows in *Fig.* 17. If the contactor is to remain unresponsive to the acceleration d^2x/dt^2, the resultant of the moments of the inertia forces taken with respect to the shaft must be zero. This is indicated mathematically, after dividing all terms of the equation by the acceleration, as $W_ay_a = W_dy_d + W_ey_e$, where W_a, W_d and W_e are the weights of the components and y_a, y_d and y_e are scalar distances from the shaft to the centers of gravity of the respective components.

BALANCED MECHANISMS—MULTIPLE PIVOTS: It has been shown that a statically balanced lever or mechanism does not respond to accelerations comprising only translatory components. This static balance is sufficient for many applications. For special conditions, a mechanism may be required to remain unresponsive to acceleration having rotational components. Assume the rheostat in *Fig.* 15 to be subjected to a rotational acceleration, as a result of which it experiences rotational displacement. The inertia of the contact arm causes it to tend to remain stationary in space, and its position with respect to

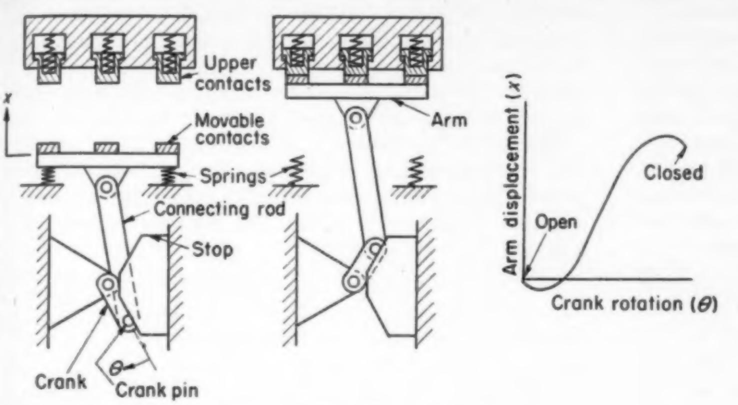


Fig. 19—Overcentering contactor shown in open and closed positions, with contact arm displacement curve shown at the right

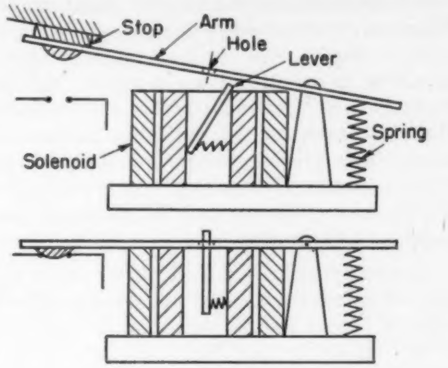


Fig. 20—Magnetic type contactor latch actuated by electromagnet in contactor

the resistance element is altered. This is characteristic of single-pivot mechanisms in general, and all of those illustrated in the preceding can be shown to be vulnerable to rotational acceleration about some axis of rotation.

Accelerations experienced by equipment generally can be resolved into (1) rotational acceleration about any arbitrarily selected axis and (2) translatory acceleration equal to the resultant acceleration of the selected axis. The requirements for a completely shock-resistant mechanism may now be formulated:

1. The mechanism must be statically balanced. Inertia forces and couples resulting from the translatory component of the acceleration are then balanced, and there is no tendency for the elements of the mechanism to be displaced.
2. The inertia forces and couples resulting from rotational acceleration about any axis must have no resultant which tends to displace elements of the mechanism. This condition can be proved by selecting any convenient axis, and computing the forces and couples acting on elements of the mechanism when the entire mechanism is accelerated about this axis. If the resultant of these forces and couples is zero, the mechanism is unresponsive to rotational acceleration about any axis, provided the requirements for static balance are satisfied.

These principles are best illustrated by an example, shown schematically in *Fig.* 18. The mechanism has two horizontally movable armatures of equal mass, which are the only elements having appreciable mass. Two bell cranks pivoted on fixed trunnions engage the armatures by means of pins. The opposite ends of the bell cranks are anchored together by means of another pin which moves in a slot in one of the bell cranks. The system is thus substantially symmetrical with respect to a vertical plane perpendicular to the paper. Since a rightward motion of one armature requires an equal leftward motion of the other armature, the mechanism is statically balanced. The mechanism thus meets the first of the two preceding requirements and is unresponsive to translatory accelerations. Its response to rotational acceleration will now be investigated.

As indicated previously, there will be no response to rotational acceleration if a single axis of rotation can be found about which the inertia forces and couples have no tendency to displace elements of the mechanism. It can be readily shown that the axis indicated in *Fig.* 18 satisfies this requirement. If the rotational acceleration of the equipment is counterclockwise, as indicated by the arrow, the inertia forces associated with the massive armatures will be directed vertically, as indicated by the arrows. Inasmuch as the armatures are constrained to move only horizontally, there is no tendency for the mechanism to be displaced by the inertia forces resulting from the rotational acceleration.

Idealization Justified in Some Mechanisms

It should be noted that the acceleration referred to in the preceding paragraph is tangential, resulting from angular acceleration about the axis of rotation. If the angular velocity should become appreciably great, centrifugal forces then act upon the armatures. These forces are directed outwardly from the axis and tend to cause displacement of the moving elements of the mechanism.

In the preceding examples, the mechanisms are idealized as elements of large mass connected by elements assumed to be massless. This idealization is reasonably justified in certain types of mechanisms. In other mechanisms, the mass is distributed and an accurate simulation of the mechanism requires that masses be assigned to all elements in order to analyze the response to accelerations. No general rule can be stated in this respect. The designer must determine, in each instance, the extent to which massive elements of the mechanism may be considered massless.

Only in special cases do shock motions include rotational components of appreciable magnitude. The majority of mechanisms can be statically balanced

to a fair degree, and therefore will be unresponsive to the predominant number of translatory shocks encountered under normal circumstances. A mechanism with a single pivot generally cannot be made unresponsive to rotational acceleration. It is necessary to employ at least two pivots for this purpose, and the mechanism almost inevitably becomes more complex. This greater complexity does not appear justifiable, unless it is known that the shock motion to which the equipment will be subjected embodies rotational components of appreciable magnitude.

OVERCENTERING: A technique which may be employed on mechanisms having fixed positions of operation is known as overcentering, consisting of setting a mechanism beyond the dead-center of a crank and against a stop. The stop prevents motion in one direction, while motion in the other direction is made improbable by the dead-center position of the crank. The three-pole contactor shown in the open and closed positions in *Fig.* 19, left and center, illustrates this principle. The movable contacts are carried by an arm constrained to move only in the vertical direction. This arm is actuated by a crank and connecting rod, the contactor being opened by counterclockwise rotation of the crank. As the crank approaches lower dead-center, the arm engages springs which are compressed upon continued downward movement of the arm and urge the crank pin against the side of the lower rigid stop when in the open position, *Fig.* 19. The contactor is closed by clockwise rotation of the crank. Upper contacts are urged downward by springs and, as the crank approaches upper dead-center, the contacts engage and the upper springs are compressed. As the crank passes dead-center, the upper springs urge the crank pin against the upper side of the rigid stop, as shown in *Fig.* 19, center.

Displacement of the contact arm is shown as a function of crank rotation in *Fig.* 19, taking as a reference the open position of the contactor shown at the left. An upwardly directed inertia force acting on the arm would tend to move the arm from open to closed position. This cannot occur directly, however, as the upward motion of the arm must be preceded by a slight downward motion until the crank passes lower dead-center. It should be noted that only a small component of the force applied to the crank through the connecting rod is in the tangential direction required to cause rotation of the crank. If the crank is statically balanced, there is only a small force tending to disturb the contactor from its open position. The same considerations can be shown to apply when the contactor is in the closed position. It should be noted that the mechanism rests against steps in its terminal positions, and that overcentering is therefore a useful technique only where the mechanism has definite operating positions. It is not applicable to mechanisms having continuously variable positions, such as the rheostat shown in *Fig.* 15.

LATCHES: Latches of various types are in the nature of expedients used where it is impossible to attian shock resistance by balancing all moving parts or by overcentering. They usually add to the complexity of the equipment; their performance, particularly those of the inertia type, generally is not reliable under a large variety of conditions.

Contactor Latch Actuated by Electromagnet

A latch may be defined as an added device which prevents accidental operation of a mechanism by obstructing the motion of one or more members of the mechanism. A well-designed latch should not interfere with normal operation of the mechanism, but should come into action when a shock occurs. Various ingenious means have been devised to prevent shock-induced maloperation of equipment.

The magnetic type of latch, as used on contactors, is actuated by the electromagnet of the contactor itself. A simple form of this device is illustrated schematically in *Fig.* 20. In normal operation the contactor is closed by energizing the solenoid. The magnetic attraction of the core upon the arm induces counterclockwise rotation of the arm against the opposing force applied by the spring, and the contact carried by the arm bridges the gap in the line. The contactor is shown in open position and in closed position in *Fig.* 20. If the contactor were not provided with a latch, it could be changed from open to closed position by an upward acceleration sufficiently large to cause a counterclockwise inertia couple acting on the statically unbalanced arm capable of overcoming the force applied by the restraining spring. The possibility of maloperation is decreased by employing a spring force as great as possible, but this spring force must be maintained small

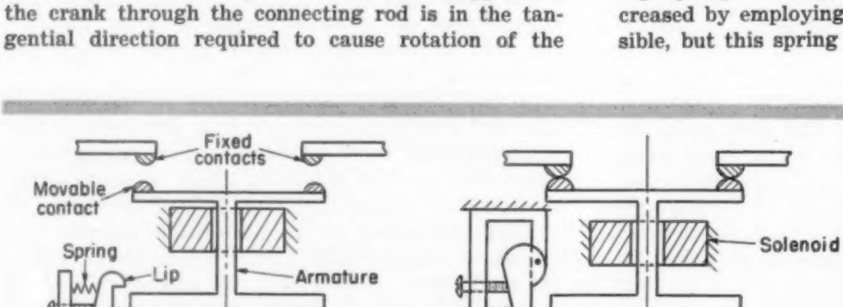


Fig. 21 — Left — Inertia actuated contactor latch. Accidental closing prevented by latch arrangement shown at (a), accidental opening prevented as at (b)

enough to be overpowered by the magnetic force during operation of the contactor. The arm is maintained within reach of the solenoid in the open position by a stop.

The latch comprises a pivoted, spring-biased iron lever located in a slot in the core. When the contactor is open, i.e., when the solenoid is not energized, a small compression spring causes the lever to remain askew with respect to the longitudinal axis of the core, as shown in *Fig.* 20, thus interfering with any counterclockwise rotation of the arm. The contactor therefore cannot be closed without first displacing the lever. When the solenoid is energized, magnetic forces cause the lever to become aligned with the axis of the core, thereby overpowering the opposing compression spring within the core. In this new position, the lever is in alignment with a hole in the arm. A *Fig.* 20 shows, the latch does not interfere with closing of the contactor when actuated by the energized solenoid. It should be noted that, while this latch offers protection against the closing of the contactor, it does not prevent opening of the contactor as a result of a downward acceleration of such magnitude that the clockwise inertia couple overcomes the magnetic force acting on the arm.

An example of an inertia-actuated latch is illustrated in conjunction with the contactor shown in *Fig.* 21 in which an armature is located within a solenoid. The contactor is shown in open position in *Fig. 21a,* with the armature urged upward toward the closed position shown at *b* when the current flows in the solenoid. The movable contacts then engage the fixed contacts and close the circuit. In the absence of a latch, the same result could be obtained from an upward inertia force acting on the armature as a result of a large downward acceleration of the contactor. When the contactor is in the closed position, maloperation could result from a large upward acceleration of such magnitude that the inertia force would overpower the magnetic force of the solenoid

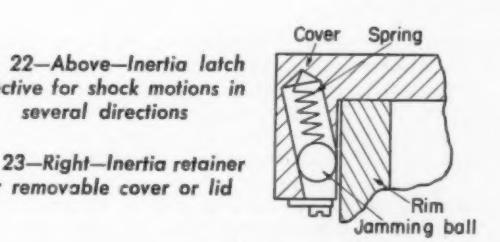


Fig. 22—Above—Inertia latch effective for shock motions in several directions

Fig. 23—Right—Inertia retainer for removable cover or lid

and open the circuit.

Closing of the circuit in response to downward acceleration of the contactor is prevented by the latch shown in *Fig. 21a*. Operation of the latch is as follows: A restraining member is pivoted at a point to the right of its center of gravity. The upward inertia force on the restraining member resulting from downward acceleration will exert a clockwise couple on this member. The latch thus rotates in a clockwise direction in opposition to the force applied by the tension spring and the projecting lip interferes with upward motion of the armature.

Accidental opening of the contactor is prevented by a similar, but inverted, latch shown in *Fig. 21b*. In this instance, an upward acceleration will result in an inertia couple causing a counterclockwise rotation of the member. A lip then prevents the armature from moving downward.

Universal Inertia Latch

An inertia latch which is effective under the influence of shock motions in various directions is illustrated in *Fig.* 22. The restraining member may function as in *Fig.* 21. The latch shown in normal position in *Fig. 22a* is urged in a counterclockwise direction by a tension spring. A chain extending leftwardly from the latch protrudes through an aperture in the fixed member and is attached to the inertia element. If the machine experiences a rightward acceleration, the inertia force acting on the inertia element separates it from the fixed member. The lower end of the latch thus moves to the left, and the lip on its upper end moves to the right in position to prevent maloperation of the associated mechanism. The latch member is statically unbalanced in such a manner that it rotates in a clockwise direction in response to a leftward acceleration. The latch is thus operative for any acceleration in a direction parallel to the chain.

The latch is also responsive to any acceleration having a component directed at right angles to the chain. This is accomplished by forming the right end of the inertia element approximately spherical with a spherical seat in the fixed member. When the machine experiences a downward acceleration, for example, the inertia force acting on the inertia element is upward and the element assumes the position shown in *Fig. 22b*. This pulls the chain through the aperture in the fixed member, and rotates the latch to a position where it is capable of intercepting the motion of associated elements of the contactor.

Considerable care should be exercised in attempting to utilize inertia latches of the types illustrated in *Figs.* 21 and 22. Two independent moving elements are employed—the armature and the latch—and success of the device depends upon always achieving a faster response of the latch than of the armature. This might be difficult to attain, considering the wide diversity of shock motions to which equipment may be subjected. Experience has shown this type of latch to be highly effective under the influence of certain shocks, but frequently ineffective when subjected to other shocks. Its use is not recommend-

ed if more suitable means can be employed to prevent maloperation.

Ingenious use is made of inertia forces to maintain the position of a removable seat-mounted member, as illustrated in *Fig.* 23. A cover rests upon the rim of an instrument, and is required to be held in position during shock but to be readily removable when desired. There are three wells in the flange of the cover, each containing a jamming ball. Under static conditions, the tension spring maintains a clearance between the ball and the rim. If the instrument experiences an upward acceleration as a result of shock, and then reverses its motion, the cover tends to part from the rim with an upward velocity. During the upward motion, however, the inertia force of the jamming ball causes the ball to overcome the spring force, and to become wedged in the tapered clearance between the cover and the rim. The cover is thus prevented from being thrown off the rim when the reversal of motion occurs. When the cover is removed slowly, the inertia forces are negligible and the position of the ball is not altered. The cover is thus readily removed.

DAMPING: While an optimum design embodies the complete balancing of all mechanisms to insure against maloperation, it often is acceptable to employ an unbalanced mechanism arranged so that maloperation is improbable. One such device is the viscous damper. This is an effective means of impeding maloperation of a mechanism but, unfortunately, is applicable only where the mechanism is not required to have a fast response. It is characteristic of such a damper that the force required to move it increases as the velocity increases. The inertia forces which cause maloperation of mechanisms are applied suddenly and tend to cause sudden motion of the elements of the mechanism. The damper resists this motion but does not materially interfere with the normal operation of a mechanism which involves low velocities of the moving elements.

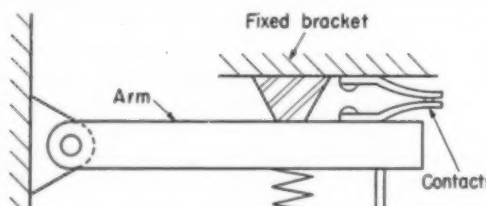
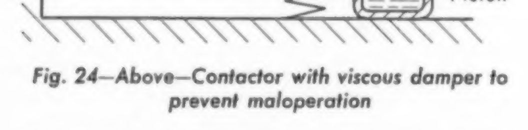


Fig. 24—Above—Contactor with viscous damper to prevent maloperation

Fig. 25—Below—Arrangement of thermostat mechanisms with conventional design shown at (a), shock-resistant mechanism shown at (b)



Damping Applied to Contactor

Application of a viscous damper to a contactor is illustrated schematically in *Fig.* 24. The pivoted arm carries at its free end a contact which engages another contact carried by the fixed bracket. The contacts are normally held in engagement as a result of the force applied by a spring. The arm is not statically balanced, and the inertia force which develops from an upward acceleration of the contactor may overpower the spring and disengage the contacts. The damper comprises a cylinder, carried by the chassis of the contactor, and a piston carried by the arm. The cylinder contains a fluid which may escape to the space above the piston either through the clearance between the piston and cylinder or through apertures in the piston. If a large downward force is suddenly applied to the arm as a result of an upward acceleration, no motion will occur immediately because the force exerted by the fluid upon the piston opposes sudden motion. The piston, however, may be moved slowly upon the application of a small force, as in normal operation. Similar results can be attained with other types of damping which exhibit a resistance to sudden motion, such as magnetic or air damping.

USE OF STIFF MEMBERS, SMALL MASSES, AND LARGE CLEARANCES: The technique of designing shock-resistant equipment calls for the balancing of all mechanisms, where feasible, and the designing of other parts so that maloperation is unlikely to occur. This latter procedure includes the use of masses as small and springs as stiff as functional requirements permit. Another important technique is the employment of mechanisms which require appreciable motion of the moving elements in order to cause maloperation. The equipment then is not likely to be affected by shocks which embody relatively little motion.

Shock resistance of pushbutton contactors can be increased by utilizing one or more of the following precautions:

1. Moving parts should be made as light as possible, consistent with required strength. Magnitude of the inertia force which tends to cause maloperation, i.e., accidental completion of the circuit, is directly proportional to the mass of the moving parts. A small mass thus decreases the tendency to maloperate.
2. Springs should be made as stiff as possible, consistent with the requisite ease of operation. The spring can be designed so that it is partially

(*Continued on Page* 184)

Graphical Analysis of Axially End-Loaded Beams

By H. D. Conway
Professor of Mechanics
Cornell University
Ithaca, N. Y.

ANALYZING members which are subjected simultaneously to longitudinal compression and lateral loading is a frequent problem in engineering practice. The usual procedure is to solve the differential equation for the elastic curve of the member and then find the values of the constants of integration by use of the conditions existing at the ends of the member.

However, such a procedure is rather laborious, especially if the lateral loading is complex; for example, if a number of concentrated loads are acting. It is therefore more convenient to make use of the following graphical method which, although not new, is apparently unfamiliar. It appears to have first been introduced by the English engineer, H. B. Howard.[1]

As an example in the use of this graphical method, a uniform compression member subjected to equal and opposite moments M_0 at its ends, *Fig.* 1, will be treated. With the origin of co-ordinates at the center of the member, the bending moment M at any section may be written in the form $M = m - Py$, where m is the bending moment due to the lateral loading and Py is that due to the end load P. By differentiation twice with respect to x,

$$\frac{d^2M}{dx^2} = \frac{d^2m}{dx^2} - P\frac{d^2y}{dx^2}$$

However d^2m/dx^2 is the intensity of loading (and therefore zero) and d^2y/dx^2 is M/EI, where E is Young's modulus for the material and I is the relevant moment of inertia of the cross-section for bending. The differential equation then becomes

$$\frac{d^2M}{dx^2} + k^2M = 0$$

where $k^2 = P/EI$. It will be observed that this equation is analogous to the one for simple harmonic motion; such a motion is the projection on a diameter of uniform circular motion. Therefore, the variation of bending moment in a member can be determined graphically by use of a circle.

The general solution of the differential equation is

[1] References are tabulated at end of article.

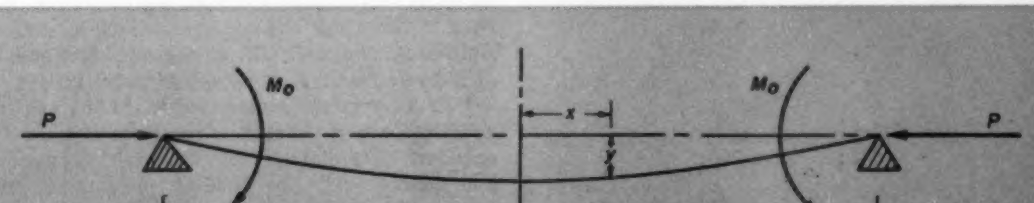


Fig. 1

$M = A \cos kx + B \sin kx$, where A and B are constants of integration. Then,

$$\text{At } x = \frac{l}{2},\ M = M_0 = A \cos \frac{kl}{2} + B \sin \frac{kl}{2}$$

$$\text{At } x = -\frac{l}{2},\ M = M_0 = A \cos \frac{kl}{2} - B \sin \frac{kl}{2}$$

Solving these equations simultaneously, $A = M_0 \sec kl/2$, $B = 0$ and hence the bending moment is

$$M = M_0 \frac{\cos kx}{\cos \frac{kl}{2}} \quad \text{(1)}$$

The shearing force V is then

$$V = \frac{dM}{dx} = -kM_0 \frac{\sin kx}{\cos \frac{kl}{2}} \quad \text{(2)}$$

The graphical construction representing Equations 1 and 2 is shown in *Fig.* 2. An ordinate AO representing the center section of the beam is erected and lines BO and CO, representing the end sections, are drawn inclined at $kl/2$ radians to the right and left respectively of AO. A point D is marked on OB so that OD represents, to any convenient scale, the bending moment M_0 at the right-hand end of the beam. A point E is marked on OC so that OE represents, to the same scale, the bending moment M_0 at the left-hand end of the beam. A circle is then drawn through the points E, O and D. Since $OD = OE$, the center O' of this circle lies on OA at the intersection of the perpendicular bisectors of OD and OE.

To find the magnitude of the bending moment and shearing force at a point distant x to the right of the center of the beam, the line OF is drawn so that the angle AOF is kx radians. If the point is distant x to the left of the center, the line OF will, of course be drawn to the left of OA.

The diameter of the circle is $OG = M_0 \sec kl/2$. Since the angle OFG lies in a semicircle, it is a right-angle and hence

$$OF = (OG) \cos kx$$

$$= M_0 \frac{\cos kx}{\cos \frac{kl}{2}}$$

which, by comparison with Equation 1, represents the bending moment at the section. Similarly,

$$FG = (OG) \sin kx$$

$$= M_0 \frac{\sin kx}{\cos \frac{kl}{2}}$$

which, by comparison with Equation 2, represents the shearing force at the section divided by the constant k. The maximum bending moment occurs at the center of the beam and is represented by $OG = M_0 \sec kl/2$. The maximum shearing forces occur at the ends of the beam and are represented by GD, at the right-hand end, and GE, at the left-hand end, multiplied by k. Thus, $V_{max} = \pm M_0 k \tan kl/2$.

It is interesting to note that when the angles BOA and COA are right angles, the lines BO and CO are colinear. The diameter of the circle becomes infinite and consequently the bending moment is infinitely large. This represents the case when P is sufficiently large to cause buckling of the member, since $kl/2 = \pi/2$ and $k^2 = P/EI = \pi^2/l^2$; therefore, $P = \pi^2 EI/l^2$, which is the Euler buckling load formula.

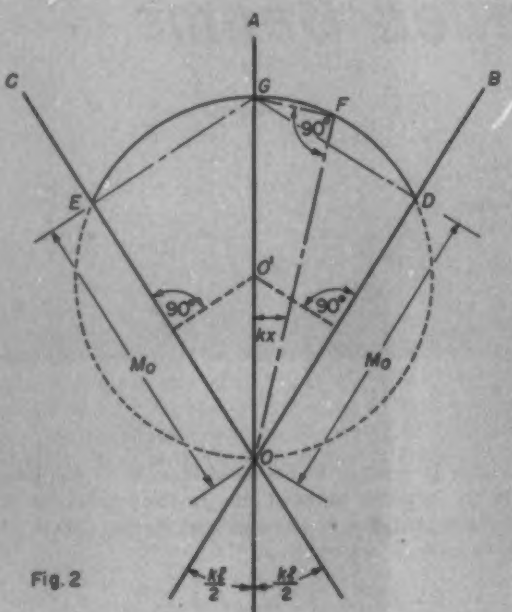


Fig. 2

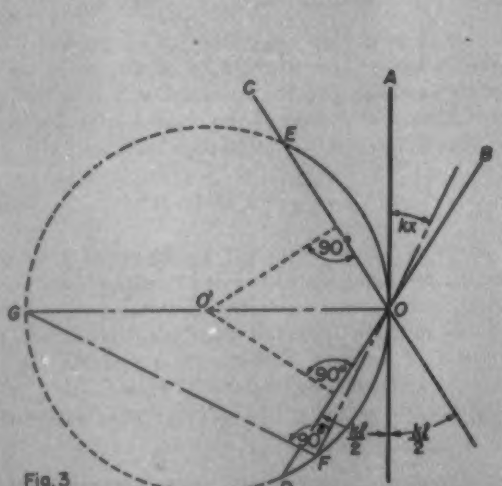


Fig. 3

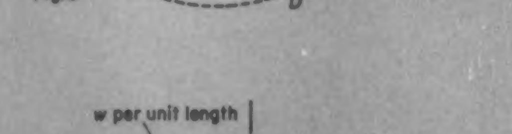


Fig. 4

If the sign of the moment M_0 at the right-hand end of the beam shown in *Fig.* 1 is reversed, the bending moment diagram is as shown in *Fig.* 3. As an example, the bending moment at a distance x to the right of the center of the member is represented by the line OF while the shearing force at the section is represented by the line GF multiplied by the constant k. It will thus be seen that the bending moment is zero at the center of the member while the shearing force there is a maximum, being equal to the diameter OG of the circle multiplied by the constant k. The bending moment also changes sign at the center section. From the figure, it may be shown that the bending moment M is given by

$$M = M_0 \frac{\sin kx}{\sin \dfrac{kl}{2}}$$

and the shearing force V by

$$V = M_0 k \frac{\cos kx}{\sin \dfrac{kl}{2}}$$

The general case of a compression member or beam subjected to any type of lateral loading may now be treated. As before, the differential equation governing the bending moment is $d^2M/dx^2 + k^2M = w$, where w is the intensity of loading. If w is constant over a certain length of the member, the solution to the differential equation for the length is $M = A \cos kx + B \sin kx + w/k^2$. If $C = \sqrt{A^2+B^2}$ and $\phi = \text{arc tan } B/A$, this equation may also be written in the form $M - w/k^2 = C \cos (kx - \phi)$. This is the equation for a circle in polar co-ordinates. The shearing force, $V = dM/dx = -kC \sin (kx-\phi)$.

General Case Analysis

As an illustration in the use of the general case equations, the member shown in *Fig.* 4 will be analyzed. In such a case, the bending moment is

$$M = \left(M_0 - \frac{w}{k^2}\right) \frac{\cos kx}{\cos \dfrac{kl}{2}} + \frac{w}{k^2} \quad \ldots\ldots\ldots\ldots\ldots (3)$$

and the shearing force is

$$V = \frac{dM}{dx} = -k\left(M_0 - \frac{w}{k^2}\right) \frac{\sin kx}{\cos \dfrac{kl}{2}} \quad \ldots\ldots\ldots\ldots (4)$$

The graphical representation of Equations 3 and 4 is shown in *Fig.* 5, the construction of which will be obvious from the figure.

The diameter OG of the circle shown in *Fig.* 5 is

$$OG = \frac{M_0 - \dfrac{w}{k^2}}{\cos \dfrac{kl}{2}}$$

The line FL represents the bending moment at a distance x to the right of the center of the member, and from the figure,

$$FL = FO + OL$$
$$= (OG) \cos kx + \frac{w}{k^2}$$
$$= \left(M_0 - \frac{w}{k^2}\right) \frac{\cos kx}{\cos \dfrac{kl}{2}} + \frac{w}{k^2}$$

as given by Equation 3. The line GF represents the

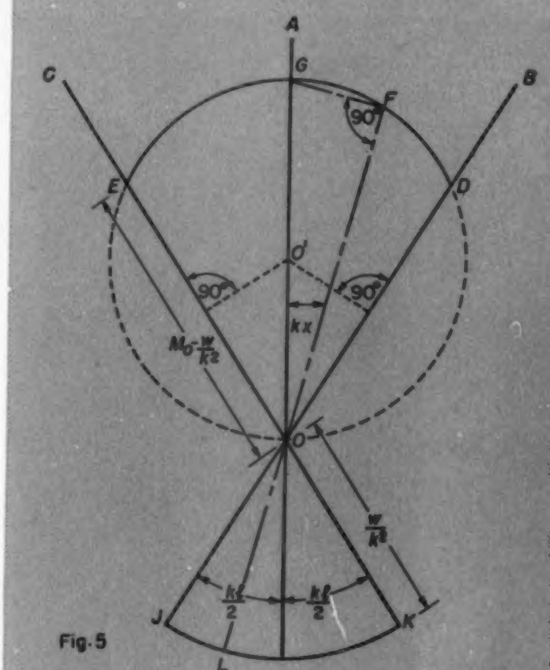


Fig. 5

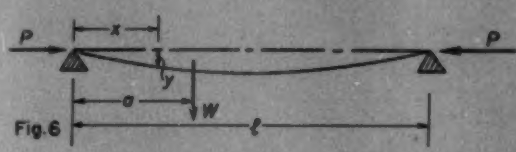


Fig. 6

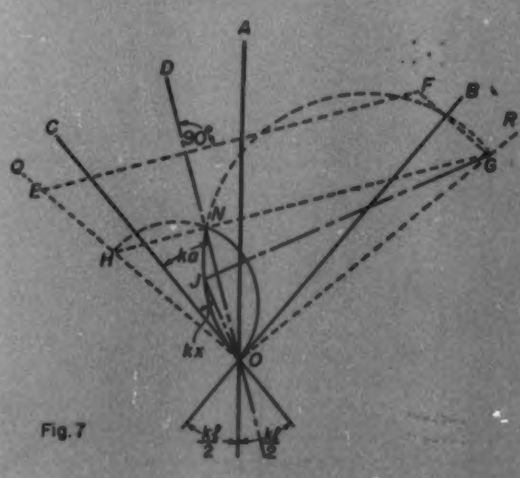


Fig. 7

shearing force at a distance x to the right of the center of the member divided by the constant k and, from the figure

$$GF = (OG)\ \sin kx$$
$$= \left(M_0 - \frac{w}{k^2}\right) \frac{\sin kx}{\cos \dfrac{kl}{2}}$$

which, when multipled by k, agrees with Equation 4. No great attention need be paid to the negative sign in Equation 4; this may be altered at will depending on the sign convention adopted.

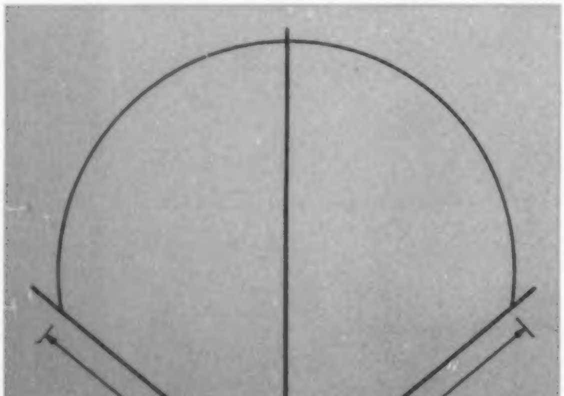


Fig. 8

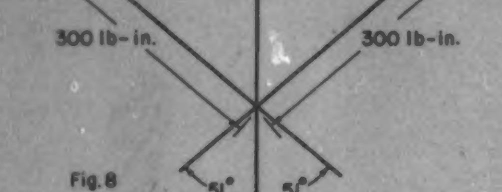


Fig. 9

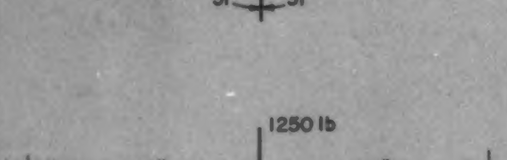


Fig. 10

As another example, the compression member carrying a concentrated lateral load W, as shown in *Fig.* 6, will be analyzed. To plot the bending moment diagram, *Fig.* 7, for this case, the lines AO, BO and CO are drawn as before to represent the center and end sections, respectively, of the member. Since the concentrated load W is a distance a from the left-hand end of the beam, the line DO is drawn such that the angle COD is ka radians; DO then represents the section immediately under the load W at which section there is, of course, a discontinuity in the bending moment and shearing force.

Lines QO and RO are then drawn perpendicular to BO and CO, respectively. From any point E on the line QO, a line EF of length W/k is drawn perpendicular to the line DO. From F, a line FG is drawn parallel to QO to intersect RO at G. The line GH is then drawn parallel to EF to intersect QO at H; GH is then equal in length to EF; i.e., W/k. To complete the figure, semicircles are drawn, with OH and OG as diameters, intersecting at N on DO.

If, for example, the magnitude of the bending moment at a distance x from the left-hand end of the member is required, the line OJ is drawn so that the angle COJ is kx radians. The line OJ then represents the bending moment at the section while JG gives, when multiplied by k, the corresponding shearing force. The bending moment under the load W is then represented by ON.

At a section immediately to the left of the load W, the shearing force is $k(NG)$ and, at a section immediately to the right, the shearing force is $k(NH)$ and is of opposite sign. The change of shearing force, as the section considered crosses the load line, is then $k(NG + NH) = k\ (W/k) = W$, which is obviously true.

Numerical Examples

The following numerical examples will serve to clarify the use of this graphical method.

Example 1: A steel tube of 1 inch outside diameter and 0.036-inch wall thickness is 20 inches long and carries a compressive load of 3000 pounds having a line of action 0.10-inch distant from the axis of the tube. Assuming $E = 30\times10^6$ psi, plot the bending moment diagram for the tube and find the maximum resultant stress.

The tube may be considered as carrying an axial load of 3000 pounds and subjected to moments of magnitude $3000\times0.1 = 300$ lb-in. at its ends. Cross-sectional area, $A = (\pi/4)(1^2 - 0.928^2) = 0.1090$ in.^{2} Moment of inertia, $I = (\pi/64)(1^4 - 0.928^4) = 0.01268$ in.^{4} Hence, $k = \sqrt{3000/(30\times10^6\times 0.01268)} = 0.0888$ in.$^{-1}$, and $kl/2 = 0.0888\times10 = 0.888$ radians $= 51$ degrees.

The bending moment diagram is then drawn as shown in *Fig.* 8. The maximum bending moment occurs at the center of the tube and is scaled to be 475 lb-in. The maximum resultant stress due to direct compression and bending is therefore of magnitude

(Concluded on Page 188)

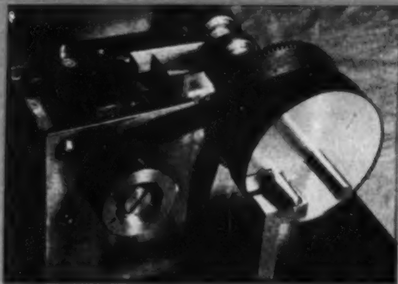
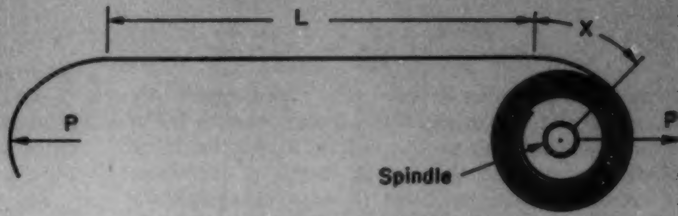


Fig. 1—Left—Type C motor in clutch mechanism giving constant torque through timing cycle

Fig. 2—Below—Recoiling force of Neg'ator depends on prestressing in length X



Prestressed Springs

. . . offer unique design characteristics

EXPERIENCE with the recently developed Neg'ator spring has proved its adaptability to unusual design applications, such as the constant torque output timing device shown in *Fig.* 1. Two outstanding characteristics of the prestressed spring are a controlled constant, positive or negative force-deflection characteristic, and extremely large deflection. These unique traits, coupled with its space-saving advantage, give the new spring made by the Hunter Spring Co. a range of applications beyonu the scope of conventional springs.

The Neg'ator is a tightly wound coil of prestressed spring material, *Fig.* 1. A force *P* is required to pull the outer end of the spring material from the coil mounted on the spindle, *Fig.* 2. In contrast to conventional springs, the force *P* does not depend upon extension, but only on intrinsic recoiling forces in the short length *X*. This force is inversely proportional to the square of the natural radius of curvature in that increment. Thus a constant or varying, positive or negative force-deflection characteristic can be produced in manufacture by controlling the incremental curvature as the spring material is prestressed. A negative force-deflection characteristic results when the natural radius of curvature increases as the outer end of the spring is extended. Conversely, a positive force-deflection characteristic will be obtained when the radius of curvature becomes progressively smaller as the outer end is extended. A constant force-deflection characteristic, or zero gradient, is produced by prestressing equal curvature along the entire length of the spring.

In present applications the unit functions in three basic forms:

1. *Extension spring*—a reacting member which tends to return to a tight coil form when the external force is removed. It consists of a single Neg'ator and one spindle
2. *Clamp*—an auxiliary clamping device consisting of a single double-coiled unit
3. *Spring motor*—a mechanism capable of storing and delivering rotational energy. It includes a single spring and two spindles.

EXTENSION SPRING: From a design viewpoint, the outstanding characteristic of the Neg'ator is a constant reacting force throughout very large extensions. Heretofore, this condition has been simulated with complicated or inefficient mechanisms such as pulleys and dead weights, helical springs and intricate cam and lever systems, or extra long and limber springs. In contrast, the zero gradient characteristic of this spring is inherent in a single part.

CLAMP: The Neg'ator in its second basic form, a clamp, is simply a spring which has been allowed to coil from both ends. The distinguishing features of the clamp are (1) constant pressure exerted between the coil heads, independent of the thickness of the clamped object, (2) automatic take-up and (3) coil separation limited only by the length of the spring stock. Because each coil head exerts force independently, the clamp may go around corners and exert pressure at different angles.

SPRING MOTOR: The Neg'ator motor is generally designed to produce constant torque at an output spindle through a large number of turns. As a source of motive power it is capable of storing energy indefinitely and delivering that energy to a system.

The two basic designs in *Fig.* 3 are designated, for obvious reasons, as type "C" and type "S" motors. In the C motor the free end of a spring is wrapped

around a spindle of radius R_2 larger than the radius of the mounting spindle. In *Fig. 3a* the radius of the mounting spindle is R_n, the natural radius of curvature of the spring. The condition of unequal forces that results causes the unit to run from the large-diameter spindle to the small one, and a counterclockwise torque is exerted on the large spindle.

Torque at the large spindle is given by the expression:

$$M = \frac{Ebt^3R_2}{24}\left(\frac{1}{R_n} - \frac{1}{R_2}\right)^2$$

where E = modulus of elasticity and b and t are the width and thickness of the spring material. The formula shows that as the radius R_2 of the large spindle is increased the output torque M increases. At extremely large radii—as R_2 approaches infinity—the C motor operation approaches that of a Neg'ator extension spring.

The C motor is primarily a low-power mechanism with an extremely long operating life. It is particularly useful as a "stand-by" or control element. For example, as an anti-backlash device, it eliminates the possibility of error by providing a constant load during excursions of recording drums and pen carriages and other similar reciprocating devices. *Fig.* 1 shows an electric clutch mechanism employing a C type motor to obtain a constant torque throughout a timing cycle. A motor winds the spring from the small to the large spindle. A solenoid then disengages a clutch, allowing the Neg'ator to run free onto the small spindle with a constant torque output.

In the S motor, *Fig. 3b*, increased output torque, with the same spindle radii as the C motor, is obtained by reverse bending the spring stock around the large spindle. Reverse bending has the same effect as increasing the diameter of the large spindle. The output torque of the S motor is

$$M = \frac{Ebt^3R_2}{24}\left(\frac{1}{R_n} + \frac{1}{R_2}\right)^2$$

As *Fig.* 4 shows, the output torque is proportional to R_2 and as R_2 increases the stress in the spring stock decreases. The decreasing stress characteristic indicates why the S motor can supply high output torques for a large number of operating cycles.

The S motor is finding more extensive application than the C type because of the added torque exerted without an increase in stock dimensions. *Fig.* 5 illustrates the design advantages of the S motor as compared to the conventional power spring motor. Zones 1 and 2 under the torque-revolutions curve for the conventional spring motor is wasted energy; no useful work is done. Insufficient torque in Zone 1 cannot actuate the driven mechanism while in Zone 2 a large amount of the total energy is not being utilized. In a Neg'ator motor with the same amount of stock, however, the part of the total energy in each motor that is wasted by the power spring motor is applied to give the constant output torque required through a larger number of turns.

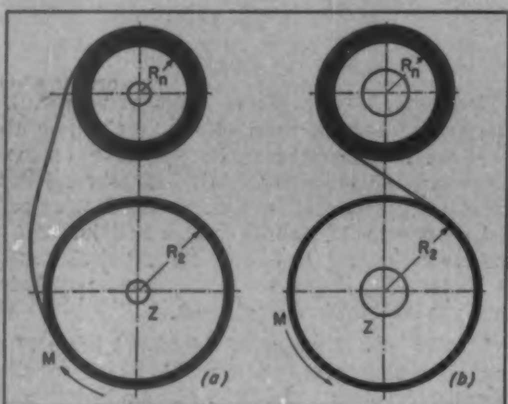


Fig. 3—Left—Diagram showing two types of Neg'ator springs, with type C shown at (a), type S shown at (b)

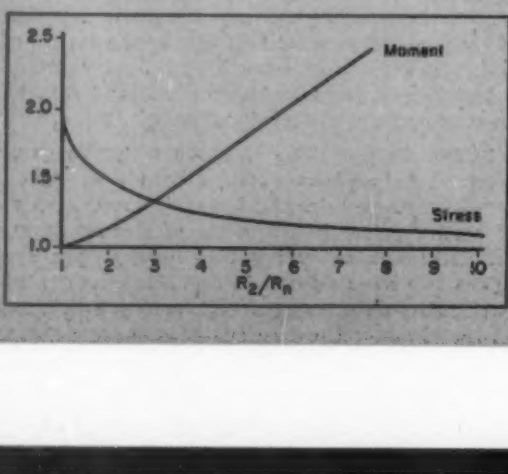


Fig. 4—Left, below—Curves showing output torque of S motor proportional to large spindle radius, and decreasing stress with increasing spindle radius

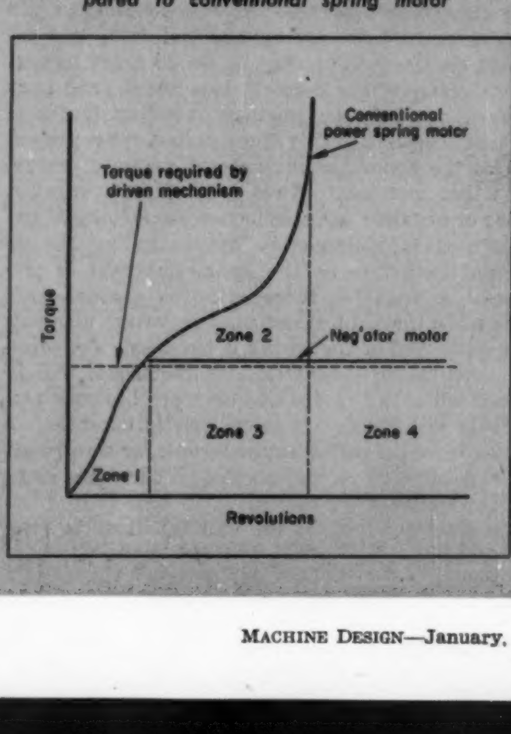


Fig. 5—Below—Design advantage of S motor compared to conventional spring motor

PRODUCTION AND DESIGN

Modern Practices in Manufacture

Costs Set by Drawing Specifications

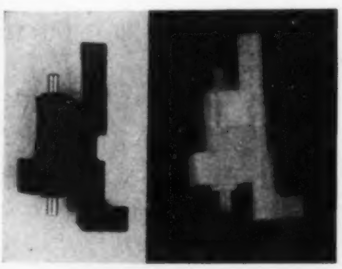
—Through Processes

Molded Plastic Eliminates Insert

THE pawl of a telephone dial transmits the motion of the finger wheel to the electrical switch which sends out dial impulses to the central office. Because of the high speed and frequent use of the dial in an operator's set, an accurately made and wear-resistant pawl is required. Until recently this part was designed with a molded hard-rubber body and a steel bearing pin, *Fig.* 1.

Seeking lower costs, Western Electric engineers considered the possible development of a one-piece plastic pawl, and nylon appeared to be the most promising material. Nylon has adequate strength for this

Fig. 1—Below—Molded hard-rubber pawl with steel pin insert, left, and one-piece molded nylon pawl, right, which replaced it



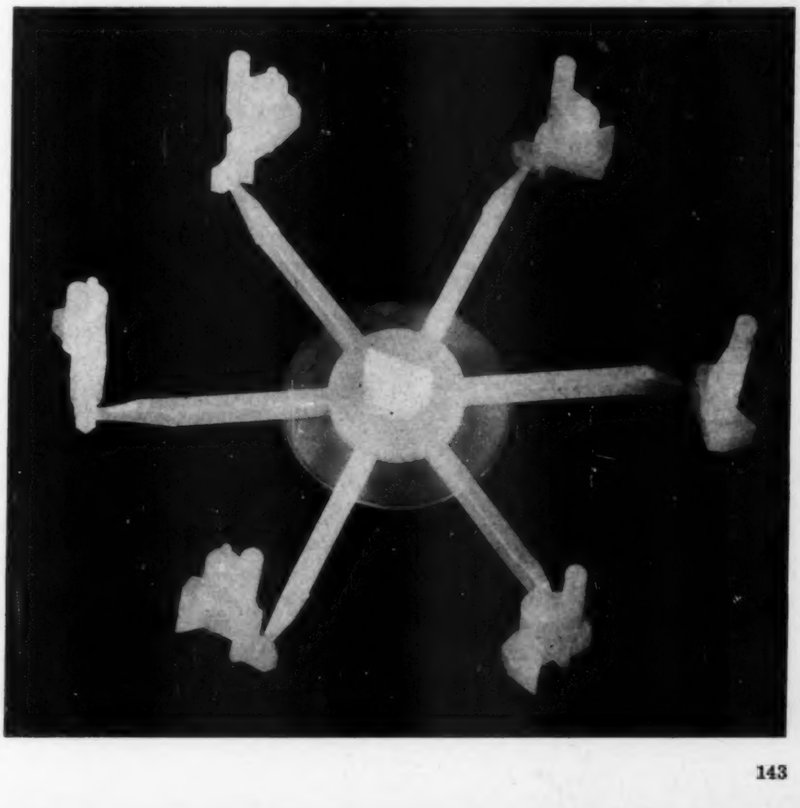


Fig. 2—Right—Group of six nylon pawls as removed from injection molding die

application, together with excellent abrasion resistance and dimensional stability and can be fabricated economically by the injection molding process. A one-piece nylon pawl was designed with only minor dimensional changes. Tests on sample parts demonstrated that it would last substantially longer than the older design. A six-cavity die, *Fig.* 2, and five-ounce injection machine are used to manufacture the nylon pawl at about one-fifth of the cost of the older design. In addition, less maintenance and better operation of the dial mechanism are obtained with the nylon pawl.

Swaging Saves Time and Material

TAPERED bobbin spindles for textile machines were formerly made on automatic lathes. Blanks 10 inches in length were turned to produce three different tapers, including a Morse taper on the shank. Following turning, the spindles were finish ground to specified dimensions.

By redesigning for swaging, material requirements are cut in half. Blanks only 4½ inches in length are needed, the material being lengthened by the swager as the tapering operations are performed. The finished length is 9 inches. Three cycles are used to produce the spindle. One set of dies swages the blank from its original diameter of ½-inch to ⅜-inch. This cycle is accomplished in 20 seconds, including loading, swaging and removal. A "bumping" operation follows with a second set of dies and takes 14 seconds. The final operation is the swaging of the Morse taper on the shank end, and this takes only 6 seconds, *Fig.* 3.

The material used is 52100 type steel having an initial hardness of 90 Rockwell B. The blanks are fed into the swager, *Fig.* 4, hydraulically and the dies are oil cooled. As previously, a finish grinding operation follows swaging, but the swaged blank, with concentricity of diameters within 0.004-inch, requires much less material removal and grinding time than does the equivalent blank which was turned on a lathe.

Physical and metallurgical characteristics of the material are improved by swaging, since a denser structure is obtained. In this respect, swaging is similar to cold forging. Alloys can be cold formed which otherwise would have to be hot forged. Too, swaging imparts a high finish to the material, often doing away with the need for special finishing operations. Distortion during subsequent heat treating is equal to or less than that resulting from turning operations.

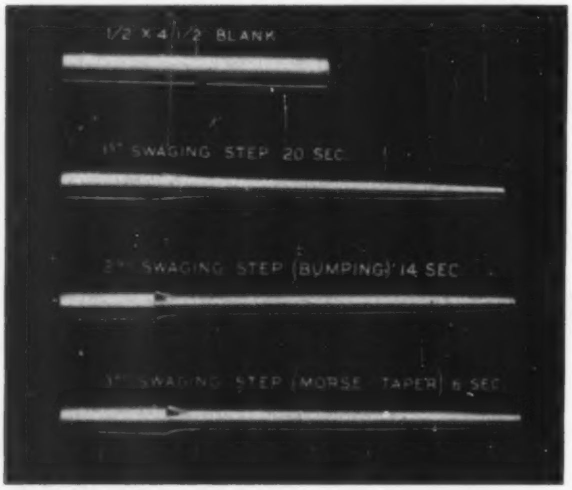


Fig. 3 — Above — Blank piece and steps in swaging to final shape

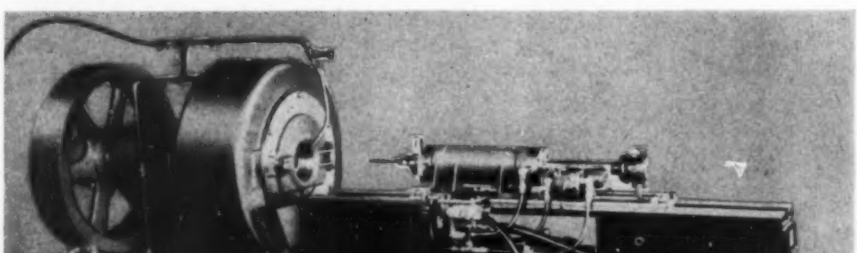


Fig. 4—Right—Fenn Mfg. Co. swager with hydraulic feed arrangement used in producing the blank shown in Fig. 3

Strain Gages In Design

Direct design results can be obtained readily by ingenious application of electric strain gages

By Sergei G. Guins
Project Engineer
Chesapeake & Ohio Railway Co.
Cleveland, O.

OF ALL the phases of engineering work, one of the most important is the experimental phase. Engineering, with all the advancements in theory is still to a large extent a matter of judgment or, as is often said, educated guessing. In the majority of cases this guessing is based on past experience, but when a new and radical design is undertaken with no past experience, the designer has a choice of going ahead with the design hoping that it will work out all right, which is often disastrous, or making a series of experiments that will show the trend if not the full answer.

The usefulness and growth of experimental engineering is closely connected with test equipment, which in turn is a product of imagination and demand of test personnel. Scratch gages, extensometers, Stresscoat, electric strain gages, and high-speed cameras all were developed by men that were faced with a problem to solve. The tools that could be used by many have become commercially available—the others can be found on the shelves of various laboratories. Each new one replaces some of the old ones but does not always make them obsolete.

It is then the duty of an experimental engineer to know what is available and how he can apply it to solve his particular problem. He also must understand the underlying principles and the function of the

Fig. 1—Top Page—Strain gages mounted on modified cantilever beam to measure large deflections. Movements of railroad axle are being recorded

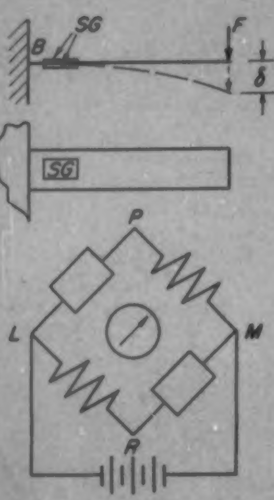


Fig. 2—Left—How large motions can be measured with strain gages and wiring diagram employed

Fig. 3 — Right — Cantilever beam mounted on railway truck for measurement of relative motions

product he is to test if his tests are to have any meaning.

As machines become more complex, the techniques and test requirements become more exacting, yet the concepts themselves never need be beyond those of elementary physics. One of the most useful tools is the electric strain gage which is used extensively for stress analysis, but if combined with the principle of Wheatstone's bridge can be used for measurements of linear and angular motions, *Fig.* 1, pressures and many other variables.

The principles underlying the application of strain gages for measurements are simple. The strain gage measures small displacements by change of resistance as the wire cross section changes. The gages are very sensitive and the deflection must be small. A Wheatstone bridge, in which one or more legs are strain gages, is used to measure voltage variations in the strain gages and, by proper calibration, the strain in the elements.

This arrangement provides tools to measure a great variety of things. *Fig.* 2 shows how large motions can be measured by strain gages. A cantilever beam having strain gages mounted on both top and bottom surfaces and connected as shown on the wiring diagram, transforms a large deflection δ, produced by force F acting on the end of the beam, into a small strain in the strain gage due to resultant tension and compression on the surfaces of the beam. The knowledge of fixity of cantilever beam connection is not necessary, as a calibration of such device, by deflecting end A a given amount and measuring the resultant change on the meter, takes that into consideration.

Use of strain gages for fixed legs of the bridge simplifies balancing of the complete bridge. The voltage across the bridge is controlled by the resistance values of the bridge and is usually 5 to 15 volts. By applying force F, the resistance of the strain gage at end B will change so that the galvanometer in the bridge will give a direct reading, due to change of potential between points P and R. If the beam has motion in two planes but values in one plane only are of interest to the investigator, a knife edge along which the beam can slide solves the problem. For dynamic work, instead of a galvanometer between points P and R a recording oscillograph can be inserted and a continuous record of motion can be obtained.

Fig. 4—Below—Heiland oscillograph used for recording movements

Fig. 5—Opposite Page—Record of movements obtained on railway truck with lateral restraint of the axle

Fig. 3 shows an installation where motion of a railroad truck was studied. Of interest were lateral motion of the axle relative to the journal box and that of the journal box relative to the frame. In the case of the axle, which is rotating, a round-headed pin was screwed into the center of the axle so that rotating friction was minimized. A knife edge was used as previously mentioned in measuring motions between the journal box and frame to record lateral motion only. The motions were recorded with the help of a Heiland oscillograph, *Fig.* 4, a six-channel instrument operated from batteries contained in the instrument. *Fig.* 5 gives a sample of a record obtained with the described instrumentation.

It must be pointed out that for installations such as the foregoing, the care with which strain gages are used can be reduced to a minimum without losing accuracy of results. In some cases tests were made an hour after strain gages were applied, the cement having been dried by infrared bulb.

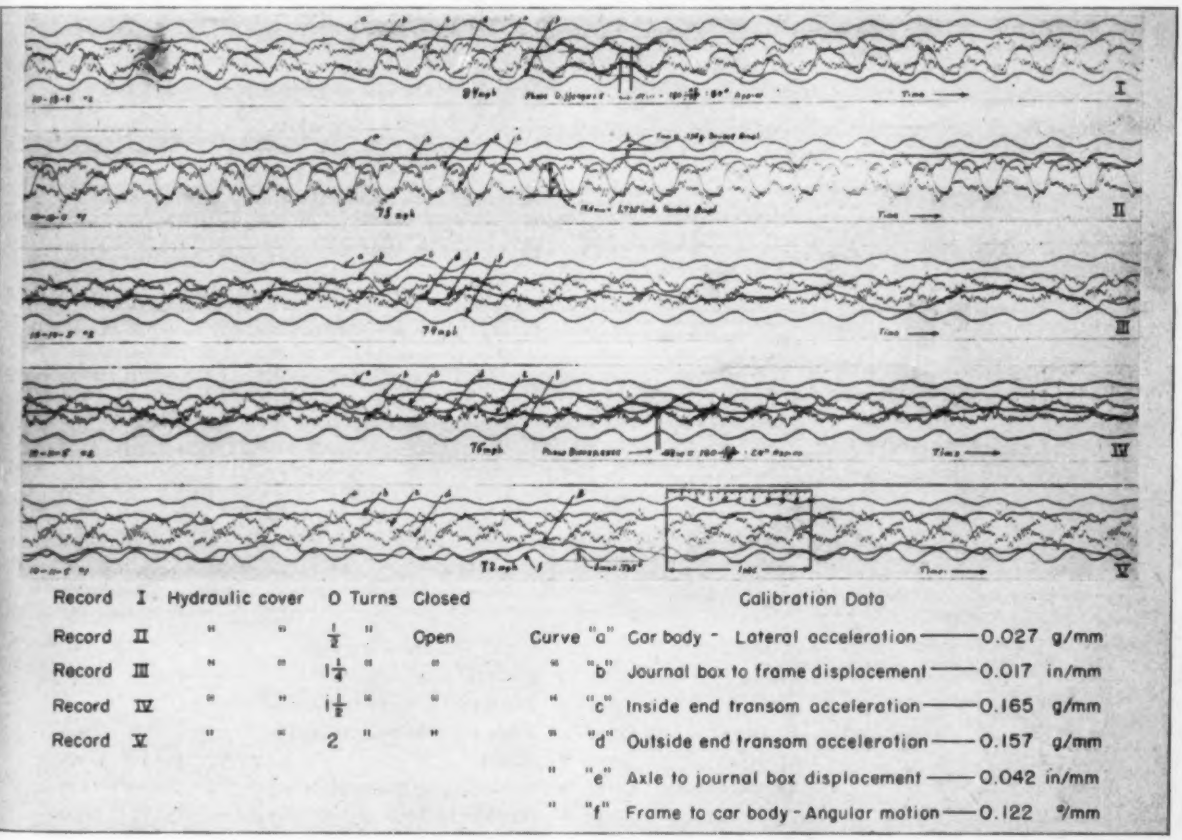
Under some conditions other variations of the beam setup can be useful. One would be to use a regular radio potentiometer as one leg of the bridge arranged so that it is rotated by a moving member, *Fig. 6a.* With this setup higher voltages can be used to gain sensitivity. Another approach is to use a slide-wire rheostat, *Fig. 6b,* which is very helpful where large motions must be measured.

Another application of a cantilever beam with strain gages, shown in *Fig.* 7, is used to measure pressure. The motion of the diaphragm under pressure displaces the beam, thus indicating pressure changes. This can be used to record pressure variations in the lines of pneumatic and hydraulic systems.

The question now arises, how can this tool I now have be applied most effectively? A question of this type is difficult to answer without knowing the problem, but a case history shows how problems have been solved by using the described instrumentation.

Increase in passenger train speeds presented a new problem to the railroads, that of increased lateral vibration in the cars. The reports indicated that vibrations were quite violent and appeared at high speeds. Further checks indicated that this vibration appeared only after the wheels of the cars had been in operation for over 30,000 miles and that changes in various components of the truck produced no improvements in the ride. An analysis of the suspension as to its natural frequencies discouraged any attempt at a theoretical analysis as there are at least nine degrees of freedom and damping characteristics are in most cases unknown.

It was then decided to approach the problem experimentally. To be able to analyze the sequence of events, measurements of motions of various components had to be recorded and, depending on conditions, accelerometers and displacement pick-ups were utilized. At the outset interest was in phase relationship of various movements and the fact that acceleration and displacement are 180 degrees out of phase had to be taken into consideration. When an instrument is selected to do a job, not only the value to be meas-



ured must be considered, but all the other conditions at the point where it will be utilized must be checked. For instance, journal box vertical accelerations are as high as 50g while longitudinal and lateral accelerations are less than 1g. If an accelerometer is used in this location it must withstand 50g, but in measuring lateral acceleration, sensitivity would be very low. On the other hand, a displacement pick-up as shown in *Fig.* 1 wouldn't be sensitive to vertical accelerations and can be made very sensitive to lateral displacements. If it is desired to have all data in units of acceleration, a simple graphical transformation can be made to change displacements into accelerations or vice versa.

The preliminary test setup was very complex, with

(*Concluded on Page* 186)

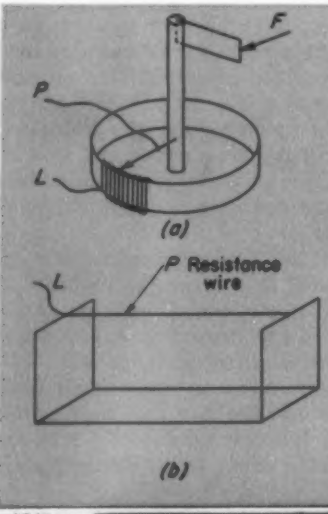
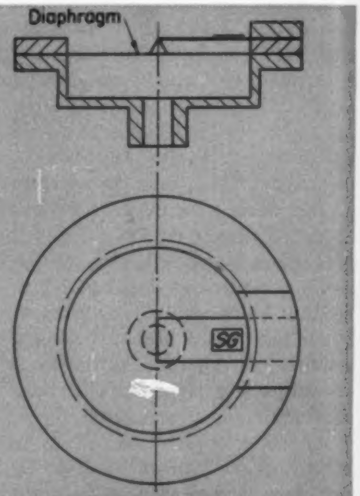


Fig. 6—Left—Potentiometer and resistance wire devices for greater sensitivity in measurement and increased range

Fig. 7—Right—Strain gage arrangement for measuring pressure variations in fluids

Fig. 8—Below—Record subsequent to that of Fig. 5 showing the improvement obtained by using angular damping between a body and truck frame



	Houde Shock Setting	Hyd. Cover Setting	Curve	Calibration Data		
Record I	Shocks disconnected	Standard covers	Curve "a"	Car body - Lateral acceleration	0.027	g/mm
Record II	Setting "b"-Very soft	" "	Curve "b"	Journal box to frame displacement	0.017	in/mm
Record III	" "d"-Very hard	$1\frac{1}{2}$ Turns	Curve "c"	Inside end transom acceleration	0.165	g/mm
Record IV	" "d" " "	$1\frac{1}{4}$ "	Curve "d"	Outside " " "	0.157	g/mm
Record V	" "d" " "	Standard covers	Curve "e"	Axle to journal box displacement	0.042	in/mm
Record VI	" "c"-Hard	$1\frac{1}{2}$ Turns	Curve "f"	Frame to car body - Angular motion	0.122	°/mm

Designing Spur and Helical Gears for Durability

By Harold M. Durham
Pittsburgh, Pa.

GEAR teeth are usually designed to meet two principal criteria: beam strength and wear resistance. For beam strength the Lewis formula is used generally, being especially valuable in applications where the pitch line velocity is comparatively low. For higher-speed gears, however, the dominant criterion usually is wear resistance. This data sheet presents a simplified method of calculating gears for wear or durability and is based upon earlier work by W. P. Schmitter.*

Capacity of a pinion to withstand wear is dependent upon pitch line load, W (pounds), pitch diameter, D_P (inches), and face width, F (inches). In simplest terms, wear resistance is proportional to FD_P/W. A more convenient index is the reciprocal of this factor, W/FD_P. At one time, a rule of thumb set the requirement at $W/FD_P = 100$. This relationship, when modified to suit average conditions, should be $W/FD_P = 60$ to 80.

Ratio of face width to pitch diameter, F/D_P, is usually assumed for trial and may range from ½ to 2. Ratios larger than 2 are objectionable because of torsional deflection.

With power input to the pinion, P (horsepower), and pinion speed, N (rpm), details can be combined in a simple equation for pitch diameters involving the factors W/FD_P and F/D_P.

The index factor is first altered to a more convenient form:

$$\frac{W}{FD_P} = \frac{W}{\left(\frac{F}{D_P}\right) D_P^2} = \frac{WD_P}{\left(\frac{F}{D_P}\right) D_P^3}$$

Power and speed are then introduced by the fundamental relationship,

$$WD_P = \frac{(12)\,(33{,}000)\,P}{\pi N} = \frac{126{,}050P}{N}$$

By substitution

$$\frac{W}{FD_P} = \frac{126{,}050P}{\left(\frac{F}{D_P}\right) D_P^3 N}$$

Finally

$$D_P = \left(\frac{126{,}050}{\left(\frac{W}{FD_P}\right)\left(\frac{F}{D_P}\right)}\right)^{\frac{1}{3}} \left(\frac{P}{N}\right)^{\frac{1}{3}}$$

$$= C_1 C_2$$

where C_1 equals the first quantity, C_2 the second.

Thus, calculation of pitch diameter, D_P, is reduced to multiplication of two factors. Factor C_1, based upon starting assumptions for W/FD_P and F/D_P, is given quickly by the chart in *Fig.* 1. Factor C_2, based upon power, P, and speed, N, can be obtained quickly by a simple slide rule operation.

An example will show how the first calculation stage is completed. Given: $P = 100$ hp, $N = 420$ rpm. Assume, as average values, $W/FD_P = 70$ and

* W. P. Schmitter—"Determining Capacity of Helical and Herringbone Gearing," MACHINE DESIGN, June and July, 1934.

Gears

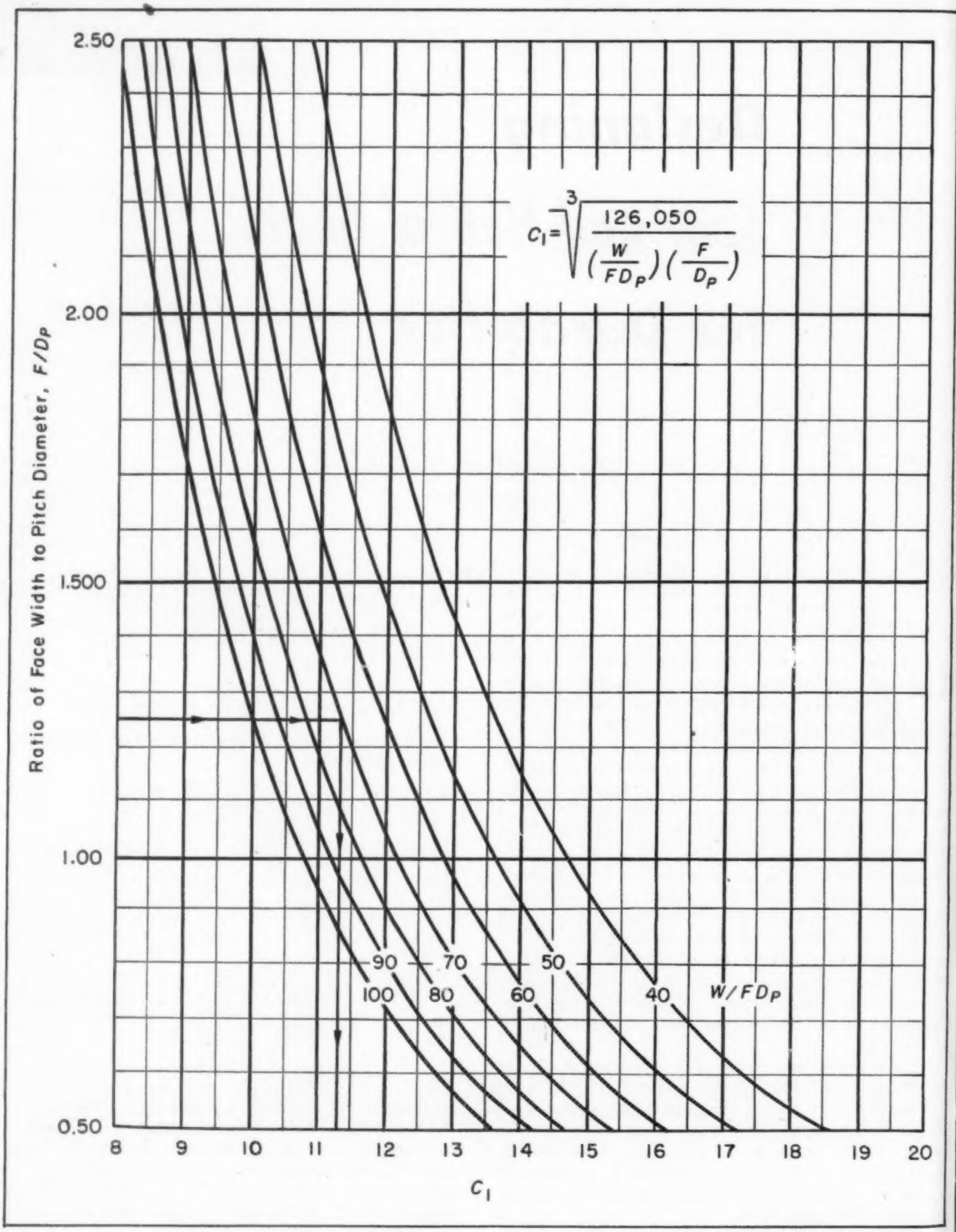


Fig. 1—Calculation factor (C_1) is dependent upon assumed values of (W/FD_P) and ratio of face width to diameter (F/D_P)

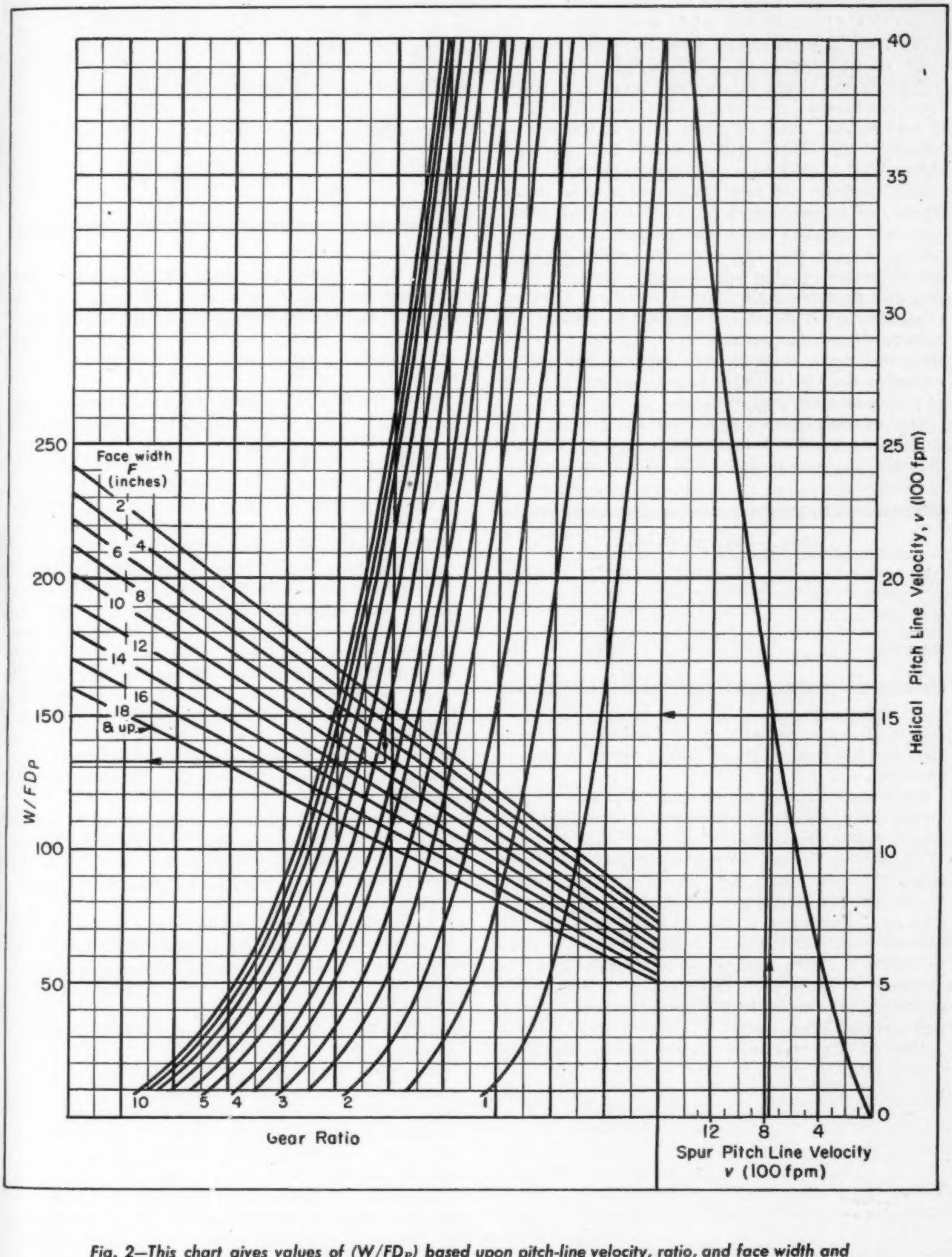


Fig. 2—This chart gives values of (W/FD_P) based upon pitch-line velocity, ratio, and face width and an arbitrary pinion hardness of 240 bhn. Table 2 gives conversion data for other hardness values

$F/D_P = 1\frac{1}{4}$. Then use of these two values in *Fig.* 1 shows that $C_1 = 11.3$. For the other factor, $C_2 = (P/N)^{1/3} = (100/420)^{1/3} = 0.62$. Pitch diameter, $D_P = C_1C_2 = (11.3)\ (0.62) = 7$ inches. Face width, $F = (F/D_P)D_P = (1\frac{1}{4})\ (7) = 8\frac{3}{4}$ inches.

Velocity and load factors may then be checked. Pitch line velocity, $v = \pi D_P N/12 = (\pi)\ (7)\ (420)/12 = 770$ fpm. Pitch line load, $W = 33{,}000\ P/v = (33{,}000)\ (100)/770 = 4290$ pounds. Load per inch of face $W/F = 4290/8\frac{3}{4} = 490$ pounds per inch.

With diameter and face dimensions now set, tooth details can be determined. Desirable contact conditions with ordinary ratios and minimizing of under cutting indicate that the minimum number of teeth for continuous running sets should be 21 teeth for a 20-degree pressure angle. Hence, for $D_P = 7$ inches, a diametral pitch, P_d, of 3 which gives 21 teeth, would be satisfactory; alternatively $P_d = 4$, giving 28 teeth. Since the choice may depend upon the selection of centers or the availability of cutters, several such sets may be calculated for convenience.

Kind of teeth depends upon pitch line velocity. For the velocity in this example, 770 fpm, spur teeth are adequate and may be full depth or stub and preferably of 20-degree pressure angle. For velocities above 1200 to 1500 fpm, single or double helical gears are preferred.

Table 1—Service Factors

Load	8–10 Hr Per Day	24 Hr Per Day	Intermittent
Uniform	1.0	1.25	0.8
Light shock	1.25	1.50	1.0
Medium shock	1.5	1.75	1.25
Heavy shock	1.75	2.00	1.50

For a more detailed study, such factors as the following must be taken into account: (1) velocity, (2) inbuilt factor (a function of size and other factors), (3) material factor (Brinell hardness), (4) gear ratio, (5) tooth type (spur or helical), (6) service factor, and (7) lubrication. The first five of these factors are incorporated in the chart of *Fig.* 2 and the conversion data of TABLE 2. Service factors are given in TABLE 1. With mineral oil lubrication, spray or pressure, a factor of 1 is employed usually. This factor may be reduced to 0.75-0.80 when extreme pressure compounds are used.

How to introduce these factors can be best illustrated by an extension of the earlier example. Let it be assumed that the gear ratio is 5 to 1, service is 24 hours per day with light shock load, and mineral oil spray lubrication is employed. From TABLE 1 the service factor is found to be 1.5; the lubrication factor is 1.

Table 2—Hardness Conversion Factors

Brinell Hardness Number	Helical Gears	Spur Gears
180	0.720	0.576
190	0.765	0.612
200	0.812	0.650
210	0.859	0.687
220	0.905	0.724
230	0.952	0.762
240	1.000	0.800
250	1.043	0.835
260	1.090	0.872
270	1.134	0.907
280	1.177	0.942
290	1.217	0.974
300	1.260	1.008

Then, the original value $W/FD_P = 70$ must be corrected by the service and lubrication factors: $W/FD_P = (70)\ (1.5)\ (1) = 105$.

Next, use the chart in *Fig.* 2 as the first step toward finding the required pinion hardness. Enter the chart at 770 fpm (spur-gear scale) and proceed along the path noted, finding an uncompensated value for $W/FD_P = 132$. This factor is actually based upon conditions for a helical pinion of 240 bhn. Obtain a correction factor by dividing the actual W/FD_P value by the uncompensated value, $105/132 = 0.795$. In TABLE 2 under "spur", note that the factor 0.795 lies between 230 and 240 bhn. Therefore, specify 240 bhn for the pinion. Since the mating gear should have a minimum hardness 30 to 40 points less than the pinion, specify 210 bhn for the gear.

The same procedure is applicable for helical pinions except that the velocity scale for helical pinions in *Fig.* 2 and the "helical" column in TABLE 2 are employed.

By this method, hardness is the unknown factor calculated from assumed initial conditions. If the calculated hardness is unsuitable for any reason, the starting assumptions must be revised and the procedure repeated.

Several limitations in the use of this method should be kept in mind. Material factors exceeding the upper limits of TABLE 2 are special with respect to hardness, leading usually to case or flame hardening. *Fig.* 2 is applicable to first reduction sets only. Additionally, *Fig.* 2 is suitable only for a 20-degree pressure angle.

DESIGN ABSTRACTS

Friction for Transient Conditions

By G. G. Gould
Naval Ordnance Laboratory
White Oak, Silver Spring, Md.

THE coefficient of friction between two materials is tacitly assumed to be a constant in most engineering applications wherein friction is encountered. Although at times such an assumption is an adequate representation of the behavior of friction forces, more often the assumption is necessitated by lack of information on friction behavior during transient conditions. For example, when a brake shoe is applied to the wheel of a locomotive, conditions do not remain constant; at the friction surface the temperature will change, the force exerted by the brake shoe will change, and the relative speed between wheel and shoe will change. As a consequence of these changes, the coefficient of friction, which is normally defined as the ratio of tangential to normal forces, also will change.

In many studies of friction, tests are so conducted that the friction material is always presented with a chemically clean, physically smooth surface by causing the friction material to travel only on previously untouched surfaces of the mating material. Such conditions are rarely met in normal applications. More frequently, the brake shoe or clutch lining operates against a metal face on which it has previously run. Thus there is the attendant surface film or surface scoring, resulting from such previous runs. The value of the coefficient of friction as published for fresh surfaces is not necessarily the same as that obtained for oxide-film-coated or scored surfaces.

In the design of a centrifugal friction clutch used for converting a variable-speed source to a constant-speed generator drive, it was necessary to choose materials for the clutch which exhibited little change in friction during 20 minutes of operation. Similar requirements are imposed by many servomechanism controls, or in the design of brakes for airplanes and trains. In these applications the designer is confronted with the evaluation of friction forces during a transient period when the friction does not remain constant. Once steady-state conditions have been established, it may be feasible to specify the coefficient of friction from "handbook" values, but this cannot be done for applications of short duration.

To enable the proper selection of materials for the clutch, experiments were performed wherein the coefficient of friction was determined as a function of time for different speeds, pressures, and temperatures. Only dry or "coulomb" sliding-friction tests were conducted. Many different friction materials were tried against many different mating materials. Chief attention was focused on the powder-metallurgy sintered materials commonly used in industry for brake and clutch applications.

It was found that for transient conditions, the coefficient of friction cannot be expressed as a simple, nondimensional number. For any two mating materials the coefficient of friction, or the force available for doing work, varies with time during the early part of a run, even under constant conditions. Furthermore, it also depends upon the speed, temperature, and pressure applied to the surfaces.

For the typical mechanical clutch or brake application, a desirable combination of friction material and mating surface is that which exhibits the following properties:

1. High value of coefficient of friction
2. Friction which is independent, or nearly independent, of pressure, relative speed between materials, and temperature
3. Low wear rate of both materials
4. No grabbing, chattering, or scoring.

Some of the observed characteristics of sliding friction between two materials are the following:

Temperature of the friction surface has a marked effect on friction. It is possible to increase or decrease the friction, even though all other factors remain constant, by increasing or decreasing the temperature.

Where constant friction is desired throughout a run, it may be more desirable to choose mating materials which will permit a very rapid surface temperature rise rather than attempt to cool the surface. In this manner, temperature equilibrium and steady-state friction may be reached more rapidly.

The powder-metallurgy friction materials exhibit higher friction normally than carbons, but also much greater wear.

Good reproducibility is not a characteristic of slid-

ing friction. Under identical conditions, on repeat runs, also at times during a given run, the friction may change unaccountably by as much as 10 per cent.

A combination of friction material and mating surface which meets the desirable conditions listed earlier is difficult to find. However, it is possible to choose materials which approach the desired characteristics of constant friction and low wear rate. For best all-around results, it was found that a hard dense chromium-plated drum with a Gempco 473 friction brush was the most satisfactory. This combination has a comparatively constant value of coefficient of friction during the first few minutes of a run; furthermore, the wear rate is comparatively low. It is worth noting that this choice of materials also yielded the best operation in the actual centrifugal-clutch application, thus substantiating the results obtained from this series of tests.

From a paper entitled "Determination of Dynamic Coefficient of Friction for Transient Conditions," presented at the ASME Annual Meeting in New York, November 26-December 1, 1950.

Designing for Investment Casting

By R. L. Wood and D. Von Ludwig
President Consultant
Arwood Precision Casting Corp.
Brooklyn, N. Y.

INVESTMENT or "precision" casting is a liquid metal-forming operation, subject to all of the variables inherent in the handling of fluid metals. Accuracy varies with the alloy used and the details of investment composition and foundry process. As a general rule those alloys of aluminum, magnesium and copper that are cast at or under 2200 F can be held to tolerances of ± 0.002-inch per inch, or part thereof of linear measurement, providing the part is well designed. In special cases one or two particular dimensions can be held to closer tolerances when the special foundry attention demanded is justified economically.

All ferrous and high-temperature nonferrous alloys must be cast to more liberal tolerances because of the character of these investment materials and greater surface roughness. A safe design limit for routine production is ± 0.005-inch per inch or part thereof. In special cases, one or two specific tolerances can be held more closely, but the designer should consult his source of supply to determine the advisability of making the attempt. In general the maximum permissible tolerances should be given for all but actual close working surfaces. When parts are correctly designed, grinding operations should suffice to finish all working surfaces, aside from the necessity for drilling, tapping, or reaming holes. Usually it is best to avoid attempting to cast small holes, unless no other way of providing them is possible.

It should be noted that castings may be produced with dimensions varying within the range of tolerances stated before. Uniformity of dimension cannot always be attained after an initial trial period, as might be expected by engineers familiar with other processes where fewer variables are encountered.

Designers frequently forget that some way of admitting the liquid metal to the casting cavity must be provided. These gates often serve the double function of feeders or risers to overcome solidification shrinkage voids. If tolerances of less than ± 0.010 inch are required at sections where gates must be ground off, special fixtures must be used at additional cost. Therefore it is desirable to have the designer specify the part or parts of the casting which may be gated without loss of functional efficiency. Where possible, gates should be against surfaces which must be finished by machining or grinding.

As a general rule not more than 0.015 inch should be specified as excess stock on any face to be machined. More is wasteful of metal and partially offsets the production advantage of the investment technique.

Thin narrow slots should be avoided. It is virtually impossible to produce slotted pocket sections open on only one side in high-temperature casting alloys. In low-temperature nonferrous alloys, such designs, though not desirable, can be produced. The proportion of length to width and thickness of each projection forming a slot determines the degree of difficulty which may be encountered in production. The mass of metal to which the projections are attached and the proportion of blending radii are also controlling factors.

Another problem of geometrical relationship is encountered in "U," "H," "T," "L," and such shapes of unbalanced nature wherein precise angularity must be attained. Liberal use of contoured fillets in the abutting sections of such parts, combined with the use of tie bars whenever possible to equalize solidification stresses, can ovecome most of the problems encountered in casting such shapes. The foundry often is forced to add ties which are then removed before shipment of the castings. It is preferable that the ties should remain until all processing has been completed.

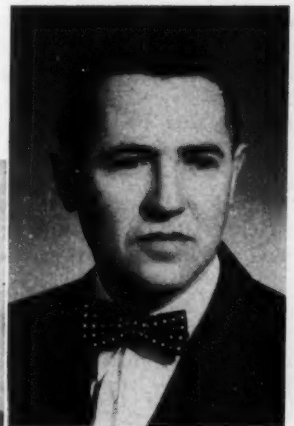
Cast Threads Should be Avoided

Except in alloys which cannot be drilled and tapped, cast threads should not be specified because, at best, only poor fits can be obtained, even with a chasing operation. Cast threads are seldom justifiable in an alloy which can be drilled; external threads should not even be specified in a hard metal, if the section can be reached and the threads imparted by grinding. Blind holes of any shape should be avoided, especially when the ratio of depth to minimum cross section is more than 3. Where through-holes are specified, it must be borne in mind that the interior surfaces will be rough, from the irregularities in the investment material, and will require at least a reaming

(*Continued on Page* 190)

MEN OF MACHINES

Charles R. Sutherland, who formerly directed engineering for smaller size Reliance motors, has been promoted to the new position of Manager of Engineering for Reliance Electric & Engineering Co.'s Ivanhoe Division products. Mr. Sutherland's new responsibilities include direction of the large motor engineering group and the high frequency engineering group. Following his first work in electric-motor design at Apex Electrical Mfg. Co., Mr. Sutherland joined the Reliance Electric & Engineering Co. in 1939 as mechanical designer. He was named mechanical engineer in charge of development in 1946 and head engineer in charge of small motors in 1947.

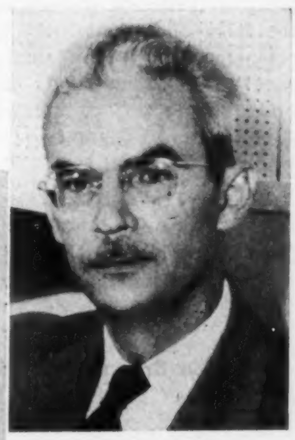


Charles R. Sutherland

William Powell Lear

William Powell Lear, chairman of the board of directors and director of research and development of Lear Inc., has been awarded the 1950 Robert J. Collier Trophy for his outstanding achievement in the development, perfection, application and production of the Lear F-5 automatic pilot and automatic approach control coupler system, which makes possible the safe landing of jet aircraft regardless of extreme weather or visibility conditions. Mr. Lear has a number of inventions to his credit starting with early automobile and home radio receivers for Motorola and Majestic as well as many firsts in the fields of aircraft radio, automatic aircraft radio direction finders, and electro-mechanical aircraft controls. He designed and perfected lightweight compact motors and actuators for the Air Force and for Lear, Inc.

Ryan Aeronautical Co. announces the appointment of **Bruce Smith** as director of engineering. Mr. Smith was formerly chief engineer of Ryan's Airplane Division. Prior to his association with Ryan in 1949, he served for nine years as chief design engineer for Consolidated Vultee Aircraft Corp. Before that, he was chief engineer for the Travelair Aircraft Corp.



Bruce Smith

Herbert J. Werner

Herbert J. Werner has joined Columbia Machinery and Engineering Corp. as chief engineer of the mechanical press division. Mr. Werner first gained prominence in the metal-working industry more than

twenty-five years ago for his work in the development of pneumatic and hydraulic die cushions while associated with the Marquette Tool and Manufacturing Co. Later, as staff engineer for the E. W. Bliss Co. and during many years as development engineer for the Clearing Machine Corp., Mr. Werner played an important part in adapting welded rolled steel construction to press design.

♦

Allis-Chalmers Mfg. Co. announces the election of **Dr. H. K. Ihrig** as vice president in charge of research.

♦

Tenney Engineering Inc. announces the appointment of several members to its engineering department. **Otis C. Wyatt Jr.**, formerly refrigeration engineer with General Electric, is in charge of refrigeration system layout. **Frank Gardner** has been appointed assistant to J. P. McCormack of the design department. **S. H. Press** and **S. B. Sternbach** have been appointed special design engineers.

♦

Frank M. Virgadamo has joined the Research Division of Burroughs Adding Machine Co. in Philadelphia as an associate research engineer.

♦

E. J. Rathsack has been appointed engineer in charge of production of the Kenworth Metal Stamping Co. Mr. Rathsack was formerly engineer in charge of the electronic induction heating department of the Allis-Chalmers Mfg. Co. Prior to that he had been vice president and chief engineer of the Industrial Electronic Corp. and in charge of production quality control and research measurements at Globe Union Inc.

♦

McCulloch Motors Corp. announces the appointment of **Walter K. Deacon** as production engineer for the company's line of lightweight engine-driven equipment and target aircraft engines. Mr. Deacon was formerly assistant to the manager of engineering and manufacturing, Marquardt Aircraft Co., Los Angeles.

♦

Murray Tribbett has been appointed chief engineer of the Hydraulic Press Division of the French Oil Mill Machinery Co. Mr. Tribbett was formerly assistant chief engineer of the Hydraulic Press Manufacturing Co.

♦

J. Calvin Brown, engineer and patent attorney, has been elected president of the American Society of Mechanical Engineers.

♦

Otto B. Blackwell has been awarded the 1950 Edison Medal for his pioneer contributions to the art of telephone transmission.

♦

Thomas C. Gray has been appointed director of engineering for Pullman-Standard Car Mfg. Co. He joined the company in 1949 as manager of engineering production.

♦

Fred I. Johnson, designer and developer of special purpose woodworking machinery, has joined the J. H. Frey Co. of Chicago. A graduate of Armour Institute of Technology, Mr. Johnson began his career in the aircraft industry, then joined the John Deere Harvester Works. He was a tool designer for the Roy E. Roth Co. and in 1943 became associated with the Aero Parts Mfg. Co. In 1945 he was appointed industrial engineer for the Rway Furniture Co. and was the company's chief engineer when he joined the Frey Co.

♦

Alexander Zeitlin has been appointed vice president of Hydropress Inc. and Loewy Construction Co.

♦

Arthur V. Bender has been appointed chief engineer, National Lead Co. Mr. Bender joined the company in 1939 as an industrial engineer for the titanium division. In 1946 he transferred to the general engineering department specializing in problems concerning titanium. In 1949, Mr. Bender was appointed supervisor of the general engineering department.

♦

Doehler-Jarvis Corp. announces the appointment of **Charles Pack** as vice president in charge of the newly formed engineering and research department.

♦

David Juelss, for the past fifteen years chief engineer, American Lead Pencil Co., has been appointed technical director responsible for design and development of new products and machinery.

♦

Ellison L. Wefel has been promoted to the vice presidency of Lombard Corp. Mr. Wefel was chief engineer.

♦

George M. Lebedeff has been appointed chief engineer of the Lenkurt Electric Co. Before joining Lenkurt Electric, Mr. Lebedeff was chief engineer at Heintz & Kaufman and an engineer with Federal Telegraph Co.

♦

R. P. Clausen, assistant chief engineer, has been appointed chief engineer of the radio tube division, Sylvania Electric Products Inc.

♦

Arthur H. Lauder has been appointed manager of engineering of the Large Motor and Generator Divisions of the General Electric Co. A graduate of the University of Wyoming, Mr. Lauder joined the General Electric Co. in 1923. In 1948 he was appointed assistant manager of the Large Motor and Generator Engineering Division and held this position until his present appointment.

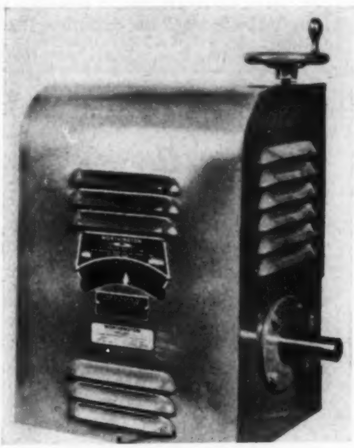
NEW PARTS

AND MATERIALS

. . . presented in quick-reference form for the convenience of the reader. For additional data on these new developments, see Page 169

Variable-Speed Transmission 1

Worthington Pump & Machinery Co., Holyoke, Mass.



Designation: Allspeed

Style: Model A, tandem belt, in-line; three frame styles, four types (output shaft above or below, left or right)

Size: To 1 hp; height 12⅞ in., width 6 13/16 in., length over shafts 13 in., shaft extensions 1¾.

Service: Light, normal, heavy or extraheavy duty; max speed variation 16:1; output range 215 to 3450 rpm at 1725 rpm input

Design: Handwheel manual or electric remote control; can be locked for constant rpm; can be direct connected, V-belt, flat belt, chain, or gear driven; transmits vertically or horizontally; life-lubricated ball bearings; belts changeable without disconnecting unit

Applications: For variable or changing reduction machine drives.

For more data circle MD 1, Page 169

Two-Way Midget Valve 2

Automatic Switch Co., 379-C Lakeside Ave., Orange, N. J.

Style: Packless; two-way, normally open normally closed; solenoid

Size: ⅛-in. pipe port with ⅛ or 1/16-in. flow holes (normally open or normally closed); ¼-in. pipe port with 3/32 or 1/16-in. flow holes (normally open); fits within 2¾-in. cube; conduit connection, ½-in. IPS; shipping weight, 1¼ lb

Service: Air, gas, water, light oil, refrigerants and other fluids to 212F; pressures—⅛-in. port 1/16-in. flow hole 300 psi (a-c) 250 psi (d-c)—⅛-in. port ⅛-in. flow hole 130 psi (a-c) 70 psi (d-c)—¼-in. port 3/32 flow hole 90 psi (a-c) 50 psi (d-c)—¼-in. port 1/16-in. flow hole 60 psi (a-c) 30 psi (d-c); 115/230 v a-c 60-cycle, 115 v d-c; power consumption, 10 watts

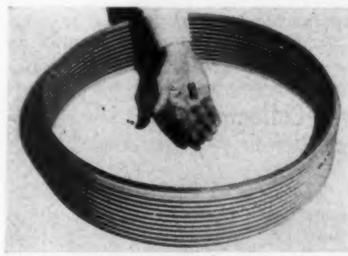
Design: Mounts in any position; body—brass or stainless steel bar stock; valve seat—crown type integral with body; valve disc—flat synthetic composition

Applications: Fluid control circuits for diaphragm motors, measuring and testing apparatus, power cylinders, etc.

For more data circle MD 2, Page 169

Stainless Steel Bellows 3

Clifford Mfg. Co., Waltham, Mass.



Designation: Hydron

Style: Flexible, seamless or welded

Size: From 5/16 to 26⅜-in. diameters with standard sizes OD x ID x wall: 7/16 x 5/16 x 0.0045; 17/32 x ⅜ x 0.0045; ¾ x ½ x 0.0045; 1⅛ x ¾ x 0.0055; 1½ x 1 x 0.0065; 2 x 1⅜ x 0.0075; 2⅞ x 2 x 0.010; 4 x 3½ x 0.014; 4½ x 4 x 0.014; 6⅞ x 5⅛ x 0.025; 15½ x 13 x 0.013; 21 x 18 x 0.021; 26 x 21½ x 0.025; wall may be varied to suit preferred wall thicknesses 0.0045, 0.0065, 0.010, 0.015 and 0.025-in.

Service: Max internal pressure at normal free length, 900 psi (5/16-in.) 50 psi (26⅜-in.); max external pressure at normal free length, 1000 psi (5/16-in.) 55 psi (26⅜-in.)

Design: Hydraulically formed; corrosion resistant; leakproof; number of convolutions as required, flexibility varies directly as the number of convolutions and spring rate varies inversely

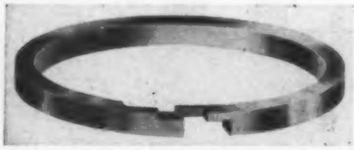
Applications: For sealing pressure or temperature control devices, flexible joints and shaft seals on mechanical equipment; larger sizes for expansion joints for jet engines and gas turbines.

For more data circle MD 3, Page 169

NEW PARTS AND MATERIALS

Piston Ring 4

Double Seal Ring Co., 2065 Montgomery, Fort Worth, Tex.



Designation: Sealock

Style: One-piece, end-locking

Size: From 6 to 22-in. diameters

Service: Two-cycle and four-cycle internal combustion engines; liner tapers to 0.010 in. per in. cylinder diameter accommodated; horsepower hours developed per gal fuel, 13.9 (plain ring) 16.4 (Sealock ring) with 0.010-in. per in. of diameter liner taper; gallons of fuel used per year on 2000-hp diesel, 865,000 (plain) 740,000 (Sealock) with 0.010-in. tapper

Design: One-piece cross section narrower than conventional rings; patented double-seal tongue and seal lock end groove

Applications: For sealing pistons in internal combustion engines.

For more data circle MD 4, Page 169

Neoprene-Base Compound 6

Stalwart Rubber Co., 180 Northfield Rd., Bedford, O.

Designation: No. 808

Form: Molded, extruded, punched or cut parts to specs

Service: Temperatures from −87 to 240F; ozone concentrations of 0.003% for 6 hours

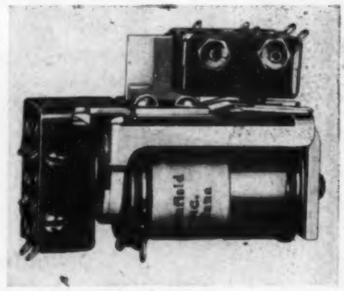
Properties: Tensile strength, 2315 psi; durometer hardness, 60; elongation, 400%; permanent set, 3%; resistant to petroleum products and their derivatives, prolonged weathering, constant flexing, and wear of metal-to-rubber contact

Applications: For grease retainers and similar synthetic parts for aircraft; also for other machine parts operating under severe conditions.

For more data circle MD 6, Page 169

Miniature Relay 8

Potter-Brumfield, Princeton, Ind.



Style: Telephone type, unhoused or hermetically sealed

Size: 1 7/16 x 1 15/16 x 11/16 in. fits 1 x 11/16 x 2 5/32 in. can

Service: 4 contacts,, 24v d-c; max inrush for ⅛-second, 12 amps; withstands 50 *g* vibration

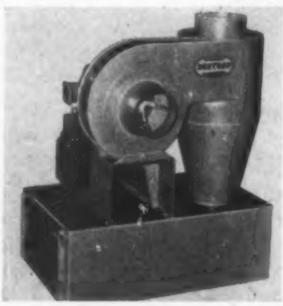
Design: 1 to 4 Microswitches bakelite enclosed; hermetically sealed has plug-in or solder terminals; actuating coils for a-c or d-c

Applications: Electrical machine relay control circuits; especially adaptable to rockets, missiles, etc.

For more data circle MD 8, Page 169

Dust Collector 5

Aget-Detroit Co., Ann Arbor, Mich.



Style: Model 8N50, filterless, portable

Size: 20 x 30-in. floor space, 5-in. diameter inlet, dust hopper 9 x 20 x 28 in.

Service: Air-entrained solids such as lint, strings from buffing, chips, shavings, sawdust, and dusts from chemicals and foodstuffs, etc.; driven by ¾-hp continuous duty motor; serves one machine; exhausts 885 cfm at 3-in. static suction

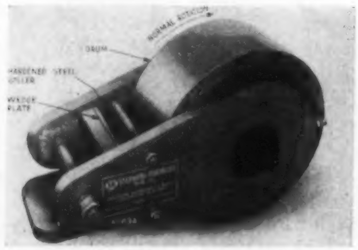
Design: Cyclone type centrifugal separation of solids; self-cleaning paddlewheel direct driven exhaust fan; air exhausted to atmosphere through outlet at top of unit

Applications: For dust removal from two grinding, buffing or polishing wheels to 12-in. diameters; woodworking machinery; multiple abrasive or sanding belts; etc.

For more data circle MD 5, Page 169

One-Way Conveyer Lock 7

Stephens-Adamson Mfg. Co., Aurora, Ill.



Style: Roller-wedge type

Size: Eleven sizes with max bores ranging 1 15/16 to 8 in. and weights 30 to 2300 lb

Servce: Max torque ranges 6000 to 500,000 in-lb

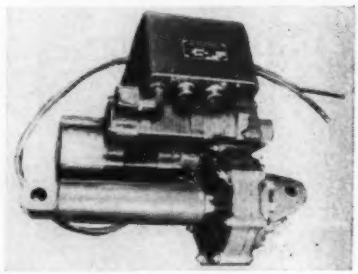
Design: Mounts on conveyer head shaft extension; steel drum keyed to rotate with headshaft, two bronze-bushed side arms hold hardened steel wedge plate at fixed distance from drum, hardened steel roller operates between drum and wedge plate; in operation, reversal of drum when head shaft stops causes roller to wedge against plate preventing rotation

Applications: For positive one-way drives on conveyors, elevators, etc.

For more data circle MD 7, Page 169

Linear Actuator 9

Electrical Engineering & Mfg Corp., Los Angeles, Calif.



Style: Dual-motor, electromechanical package

Size: 3.3 and 1/10 hp motors; unit 15 in. long retracted, 8 in. wide, 14 in. high

Service: Drive rates 0.060-in. per second and 0.700-in. per second to screw jack; normal operating load 11,000 lb, static load 80,000 lb; 28v d-c

Design: 1/10-hp motor for automatic control, 3.3-hp motor for higher rate overriding manual control; motors operate through individual gear reductions to provide the different drive rates; overload and limit switches, radio noise filter, position indicator and nonjamming stops

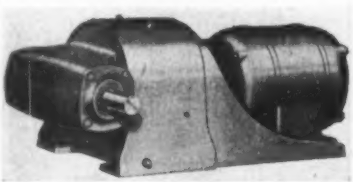
Applications: For actuation of stabilizers on turboprop and jet aircraft; screwjack actuation of units in special aircraft and industrial machinery.

For more data circle MD 9, Page 169

NEW PARTS AND MATERIALS

Right-Angle Motoreducer 10

Falk Corp., Milwaukee 8, Wis.



Style: Right-angle; All-motor for any type foot-mounted motor, integral with any D type flanged NEMA motor

Size: All-motor, 1 to 50 hp; integral, 1 to 30 hp

Service: Ratios, 500:1 min, 972:1 max; input speeds to 1750 rpm and higher if necessary

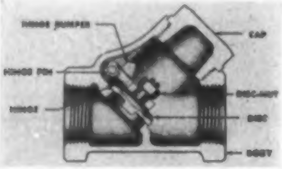
Design: Sealed all-steel housings with one-way vents; spiral bevel gears for final reduction, carburized, hardened and lapped; helical gears for high-speed and intermediate reductions, heat-treated alloy steel shaved, crown-shaved pinions; splash lubrication

Applications: For speed reduction in horizontal or vertical drives.

For more data circle MD 10, Page 169

Check Valve 11

Crane Co., Michigan Ave., Chicago 5, Ill.



Style: Y-pattern; Nos. 36 and 76E screw end or Nos. 38 and 77E flange end

Size: Screw end ¼ to 3 in. inclusive, flange end 1 to 3 in. inclusive

Service: Steam (No. 36) 200 psi 500F max, (No. 76E) 300 psi 550F max, (No. 38) 150 psi 500F max, (No. 77E) 300 psi 550F max; cold water, oil or gas (No. 36) 40 psi max nonshock, (No. 76E) 1000 psi max nonshock (up to 2-in.) 600 psi max nonshock (2½ and 3-in.), (No. 38) 225 psi max nonshock, (No. 77E) 500 psi max nonshock

Design: Hinge pin aluminum-silicon bronze, body special alloy high-pressure valve brass, other parts brass; 45-degree disk seat integral with body, positive backflow closure; valve easily reground

Applications: For checking fluids flowing in horizontal lines, vertical lines with upward flow or inclined lines with upward flow.

For more data circle MD 11, Page 169

Gasoline-Resistant Wire 12

General Electric Co., Construction Materials Dept., Bridgeport 2, Conn.

Designation: Geotrol

Form: Single-strand plastic covered; seven colors

Size: 10, 12 and 14 Awg; 44, 31 22 lb per M ft respectively; 0.19, 0.16 and 0.15-in. OD, respectively; 500 ft coil per carton

Service: UL Type TW, 600 v, 60 cycle; max operating temperature, 30 C

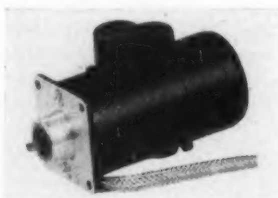
Properties: Insulated with Flamenol, resistant to petroleum products and moisture; lightweight; easily installed and stripped; replaces lead-covered wire

Applications: Used as open wiring or in recognized metal raceways exposed to gasoline or gasoline vapors such as in engine drive areas, lighting circuits, gas pump islands, etc.

For more data circle MD 12, Page 169

Small D-C Motor 13

Lear Inc., 110 Ionia Ave., NW, Grand Rapids, Mich.



Style: Model BB-05A-1, flange or base mounting

Size: 1/40-hp; 2.29 x 1.56 x 3.06 in. long; 0.75 lb

Service: 15,000 rpm; 26 v d-c; duty cycle, 3 minutes on 17 minutes off; ambient temperature range, −65 to 165 F; starting torque, 450% full-load torque

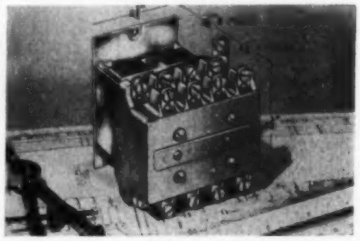
Design: Precision built, lightweight, corrosion resistant; optional features—clutch for positioning control of friction load devices, electromagnetic brake for positioning control requiring extreme holding torque or against large load inertias on motor, thermal protector for motors under conditions of maximum stress, and radio noise filter to meet ANM-40 specs

Applications: High-speed drive for aircraft and industrial equipment.

For more data circle MD 13, Page 169

Multipole Contactor 14

Arrow-Hart & Hegeman Electric Co., Hartford 6, Conn.



Style: CRA, multipole, convertible

Size: "00" 2¾-in. high, 2¾-in. wide, 3¼-in. deep (8 pole) 2¼ in. deep (4 hole)

Service: 6-600v a-c, 6-230v d-c

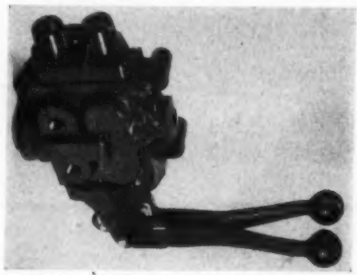
Design: Two to eight poles; four parts—frame containing magnet, coil and right-angle mechanism which permits vertical contact and horizontal make-and-break, all contacts are reversible without disassembly of contactor

Applications: For multiple electric control systems in machines.

For more data circle MD 14, Page 169

Hydraulic Control Valve 15

Parker Appliance Co., Cleveland 12, O.



Style: Open center dual-control double spool; 3-way, 4-way or combination

Size: Any port size to ¾-in. pipe

Service: Hydraulic fluids to 2000 psi; 14 gpm capacity, 20 gpm max may be handled; pressure drop through open center, 32 psi at 14 gpm

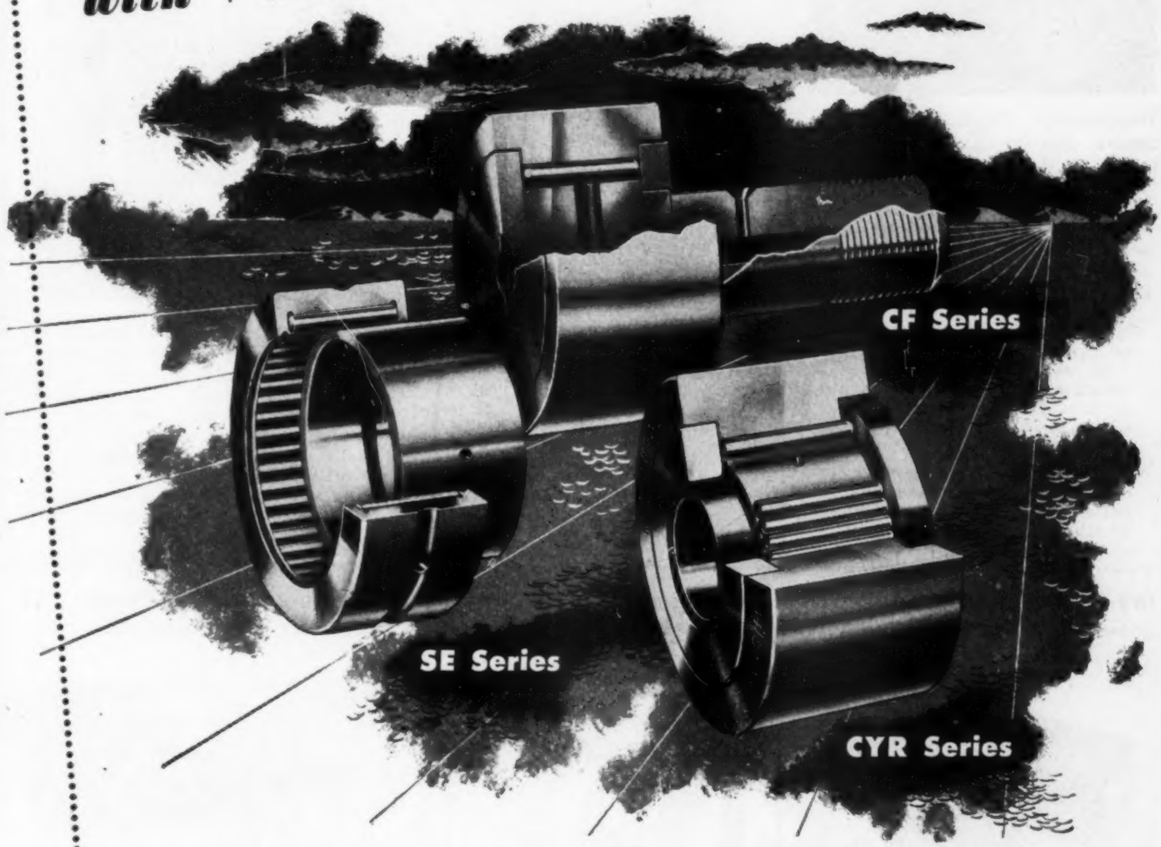
Design: Built-in balanced seatless relief valve externally adjustable, minimum pressure rise bypassing full pump capacity; double operating handles mounted at either end, upright or inverted; return port may be located for direct tank mounting; float position, in addition to regular raise, lower and hold, may be incorporated

Applications: For control in heavy-duty hydraulic circuits on farm implements, materials handling machinery, etc., especially adapted for single or double-acting cylinders not working at same time.

For more data circle MD 15, Page 169

"longer life and greater efficiency"

with McGILL® MULTIROL® Bearings



Compared with friction hampered plain bearings McGILL MULTIROL full type roller bearings resist wear because of their easily lubricated roller line contact with raceways. Longer life and greater efficiency is the natural result of the constant precision maintained in the bearing. Lubrication is simplified and required much less frequently. Maintenance is free of costly down time for periodical adjustments and bearing replacement. MULTIROL bearings are your assurance of adding these advantages to the operation of your machinery. Millions in service to date prove such exclusive features as one piece outer race and roller retaining end shoulders and lubrication reservoirs above roller ends in the SE Series. CF and CYR Series have thick outer race sections for heavy shock load in cam action applications. Mounting is possible with or without stud. Write today for Bulletins on all MULTIROL Bearing series.

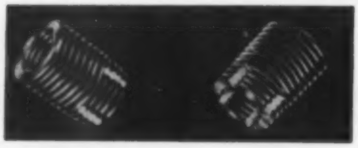
McGILL® precision bearings

McGILL MANUFACTURING CO., INC.
BEARING DIVISION, 200 N. LAFAYETTE ST.
VALPARAISO, INDIANA

NEW PARTS AND MATERIALS

Tapped Insert Fastener 16

Groov-Pin Corp., Union City, N. J.



Designation: Tap-Lok

Style: Self-tapping threaded, self-locking female tapped sleeve

Size: 11/64-40 to ⅜-20 external thread; 4-40 to ¼-20 internal; 15/64 to 31/64-in. lengths

Service: For plastics, woods and soft metals

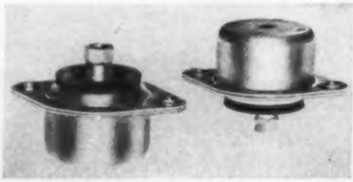
Design: Brass or case-hardened carbon steel, stainless steel on quantity order; can be driven into cored or drilled hole; taps own thread with edges of slotted segments, which are deflected radially inward to provide self-locking action on screw used and increase anchorage grip

Applications: Threaded anchorage in soft materials, for assembly of parts with machine screws, bolts or studs.

For more data circle MD 16, Page 169

Vibration Mount 18

Barry Corp., 179 Sidney St., Cambridge, Mass.



Designation: Barrymounts

Style: Series 6475 air-damped

Size: 1-in. diameter, 1-in. overall height under min rated load

Service: Load rating, 0.3 to 3.0 lb

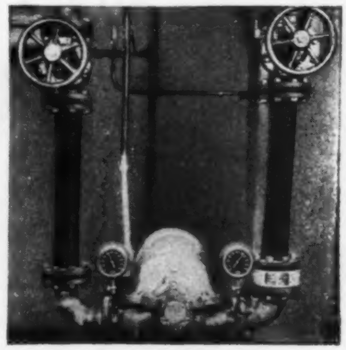
Design: Requires only ½-in. clearance; two-hole mounting on 1.414-in. centers; center stud tapped to ¼-in. depth with 8-32 thead

Applications: Vibration - isolating mounting of lightweight airborne and similar equipment installed in restricted spaces.

For more data circle MD 18, Page 169

Flexible Pipe Connector 20

Finn & Co., New York 30, N. Y.



Style: Flanged rubber coupling

Size: ½ to 14-in. pipe diameters

Service: Working pressures to 250 psi with a safety factor of 5; temperatures to 250F

Design: Rubber pipe with embedded standard metal flanges at ends

Applications: Noise and vibration absorption between pumps and pipelines.

For more data circle MD 20, Page 169

Dry-Air Compressor 17

Romec Div., Lear Inc., Elyria, O.

Style: Model RG-8160, compressor-motor unit

Size: Overall height, 4 5/16 in.; overall length, 6 31/64 in.; weight, 3.5 lb; inlet port, 1/8-in. ANPT female; outlet port, 1/8-in. ANPT male on check valve; motor, 1/15-hp

Service: Continuous duty; temperature range 55 to −55C; maintains sea-level pressure from 35,000 to 50,000-ft altitude; rated at more than 900 cim at sea level, 80 cim at 7 in. Hg absolute; input voltage, 27 v d-c; input current, 3 amps at sea level, 6 amps max at −55C for starting

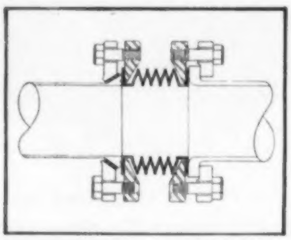
Design: Self-lubricating; packless; positive displacement pump, Graphitar blades; inlet type air filter-dehydrator; absolute pressure switch; check valve permits pump to stop running intermittently without pressure loss

Applications: For supplying oil-free compressed air to electronic units in high-altitude aircraft.

For more data circle MD 17, Page 169

Expansion Joint 19

United States Gasket Co., Camden, N. J.



Style: Flexible bellows type

Size: 1, 1½, 2, 2½, 3, 4, 6, 8, 10, and 12-in. IPS; respectively, 2, 2, 3, 3, 4, 4, 4, 4, 5, and 5-in. face-to-face width, relaxed; integral gasketed flanges to 150 lb ASME standard

Service: Corrosive fluids; pressures to 25 psi; temperatures −150 to 550 F

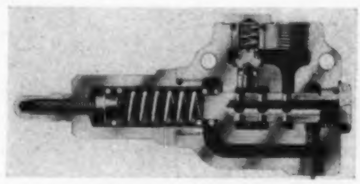
Design: Molded Teflon; chemically inert, heat resistant; gasketed flange or taper connections; longitudinal movement, ±½-in. on 1-in. IPS to +1 and −2 in. on 12-in. IPS, additional bellows sections available for increasing movement

Applications: For absorbing vibration, misalignment or thermal expansion in glass-lined, pyrex or porcelain piping assemblies, especially on chemical processing machinery.

For more data circle MD 19, Page 169

Pressure Reducing Valve 21

Hydraulic Equipment Co., Cleveland 17, O.



Style: Models VA1 and VA2, spring-loaded plunger, pressure reducing

Size: 8 9/16 in. high by 4 1/16 in. wide; reduced pressure outlet, ½-in. pipe tap; high-pressure inlet, ½-in. pipe; oil reservoir outlet, ⅝-in. pipe

Service: Oil-hydraulic fluids; max inlet pressure, 1000 psi; reduced pressure range, 100 to 300 psi (VA1), 200 to 1000 psi (VA2); capacity, 15 gpm

Design: Pressure control spring-adjusted; bypass prevents pressure buildup at cylinder port; housing cast alloy iron; plungers, heat-treated alloy steel; spring, alloy steel; other components, cold-rolled steel

Applications: For pressure reduction at any point or to any particular unit in oil-hydraulic control circuits of machines.

For more data circle MD 21, Page 169

Rubber Parts...

NEW CATALOG CONTAINS DATA ON COMPOUNDS, APPLICATIONS AND FABRICATION METHODS

This 16-page, illustrated, multi-colored catalog now is available to design, production, purchasing and management personnel. This publication has been compiled to familiarize readers with Stalwart-developed rubber compounds which feature resistance to (1) abrasion, (2) chemicals, (3) high and low temperatures, (4) petroleum products and derivatives, and (5) weathering. Sections of the catalog are devoted to the new and outstanding Silicone Rubber compounds, the major methods of fabrication, and Stalwart production facilities.

More than 60 Stalwart-developed compounds are listed by code number. Charted individually in conjunction with these compounds are their physical properties and general characteristics, as well as suggested applications.

Catalog 51SR-1 will be sent upon receipt of coupon or on letterhead request.

STALWART RUBBER COMPANY

1180 NORTHFIELD ROAD · BEDFORD, OHIO

THE STALWART RUBBER COMPANY

Please send (without obligation) your new 16-page, illustrated, multi-colored Catalog 51SR-1.

NAME ______________ TITLE ______________

COMPANY ______________

ADDRESS ______________

CITY ______________ ZONE ______ STATE ______________

NEW PARTS AND MATERIALS

Plastic Laminate 22

Formica Co., Cincinnati 32, O.

Designation: Z-80

Form: Resin-bonded coarse-weave fabric sheet; laminated-molded shapes; semigloss, natural color

Size: Minimum thickness, 1/16-in.; max 1 in.

Service: 60-cycle insulation

Properties: Arc resistance, 130 seconds; power factor as received, 0.015 (60 cycles) 0.039 (1000 Kc); dielectric constant as received, 5.10 (60 cycles) 4.80 (1000 Kc); sp. gr., 1.38; shear strength (ASTM D732-46), 14,000 psi; tensile strength, 11,000 psi (lengthwise) 6000 psi (crosswise); flexural strength, 20,000 psi (lengthwise) 14,000 (crosswise); compression strength, 40,000 psi (flatwise); Izod impact strength, 1.3-1.5 ft-lb per in. notch (flatwise) 0.8-1.0 (edgewise); water absorption, 0.6% (⅛-in. thick) 0.35% (½-in.); good punching and machining qualities

Applications: For condensers and adapters in radio-TV units; insulating power generating equipment and other 60-cycle assemblies.

For more data circle MD 22, Page 169

Flexible Coupling 24

Lord Mfg. Co., Erie, Pa.

Style: Flexible, bonded synthetic rubber

Size: Drives to ½-hp

Service: 1750 rpm; angular misalignment to 2 degrees, parallel misalignment to 1/32-in.

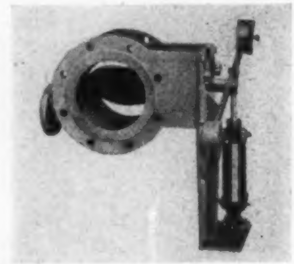
Design: Neoprene permanently bonded between flanged hubs; neoprene stressed in shear for max torsional flexibility; hubs drawn to D-shape to mate with milled flats on standard fractional-hp motors and driven shafts

Applications: For coupling fractional-horsepower drives to electrical appliances, office machines, etc.

For more data circle MD 24, Page 169

Relief Valve 26

R-S Products Corp., 4600 Germantown Ave., Philadelphia 44, Pa.



Style: No. 708, off-center disk type

Size: 8-in. diameter; 150-lb American Standard raised face flanges; smaller and larger sizes available

Service: Air, gas, liquid and semisolids; saturated steam to 15 psig with low pressure drop; units for higher pressures available

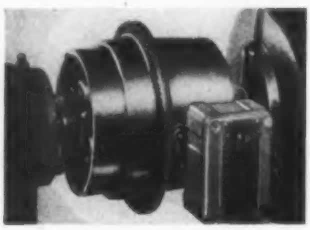
Design: Double crank arm with adjustable weights for accurate setting; cylinder dashpot; flanged outlet for overload relief

Applications: For volume control and shut-off of fluid systems on processing machinery, etc.

For more data circle MD 26, Page 169

Cutout Coupling 23

Anchor Steel and Conveyor Co., Dearborn, Mich.



Style: Flexible in-line with torque overload protection

Size: Two sizes, 3C and 5C, 7 and 8½-in. diameters respectively; 3C, 20 and 25 lb; 5C, 45 and 50 lb; max bore motor end 3C, 1¼ in., 5C, 1¾ in., max bore drive end 3C, ¾ to 1⅝ in., 5C, 1 to 2½ in.

Service: Balanced to 2400 rpm max; 3C, 0.16 to 0.24 hp per 100 rpm, 10 to 150 in-lb torque; 5C, 0.24 to 3.20 hp per 100 rpm, 150 to 2000 in-lb torque

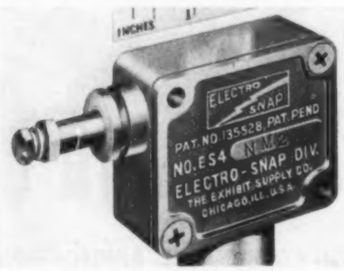
Design: Spring-loaded roller cam follower in coupling actuates grooved flange axially to contact limit switch, cutting power at rated torque, resets automatically; switches available, Micro type OP-AR, SPDT splashproof or, on request, explosionproof EX-AR similar to OP-AR

Applications: For automatic power cutoff protection against overload in any type direct-driven machine.

For more data circle MD 23, Page 169

Limit Switch 25

Electro-Snap Switch Div., Exhibit Supply Co., Chicago 1, Ill.



Style: ES4-NM, momentary, one-way action

Size: Overall length with extended plunger, 2 13/16 in.; overall height, 2¼ in.; wire outlet, ½-14 straight pipe tap; net wt, 0.30 lb

Service: Electrical capacity, UL, 10 amp, 125 or 250 v a-c; ambient temperature range, −67 to 200 F; operating force, 3½ lb; standard "on" period ⅛-in., overtravel 3/16-in., NM2; NM1 "on" 1/16-in., overtravel ¼-in. and NM3 "on" 3/16-in., overtravel ⅛-in.

Design: Pretravel, 0.050 in., movement differential, 0.022 ± 0.010 in.; wired normally open, normally closed, two-circuit, or SPDT;

Applications: For circuits requiring single pulse in control of automatic sequence movements, interlocking, positioning pulsing electric counters, initiating machine time cycles, etc.

For more data circle MD 25, Page 169

Locking Fastener 27

Shakeproof Inc., Div. of Illinois Tool Works, Chicago 39, Ill.

Designation: Keps

Style: American Standard light, machine screw or regular nuts with lock washer

Size: Light nut ¼-28, 5/16-24, ⅜-24, 7/16-20, and ½-20; machine screw nut 6-32, 8-32, 10-24, 10-32, ¼-20, and ¼-28; regular nut ¼-20, 5/16-18, ⅜-16, 7/16-14, and ½-13

Service: Average machine fastening requirements, Class 2-B threads

Design: Light nut semifinished hexagon steel, machine screw nut hexagon steel, regular nut semifinished hexagon steel; preassembled twisted-tooth lock washer swivels; funnel design of washer pilots nut onto thread

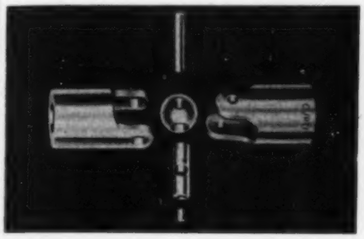
Applications: For vibration-resistant machine part assemblies; especially adapted to hopper-feed assembly techniques.

For more data circle MD 27, Page 169

NEW PARTS

Universal Joint 28

Curtis Universal Joint Co. Inc., Springfield 7, Mass.



Style: Ball type

Size: Four sizes; hub length ¾, 31/32, 1 3/16, and 1¼ in.; total length 2, 2 11/16, 3⅜, and 3¾ in. respectively; O.D. ½, ¾, 1, and 1¾ in.; bore ¼, ⅜, ½, and ⅝-in.

Service: Light duty; static torque rating 170, 650, 1250, and 2500 in.-lb; tension load 720, 2000, 3600, and 4950 lb

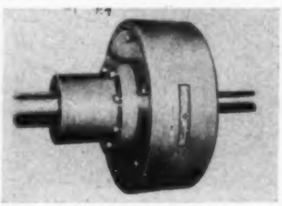
Design: Steel forks bearing on bronze ball; heat-treated centerless ground pivot bearing pins; full length intersecting bearing pins; large pin provided with oiler

Applications: Light torque drives.

For more data circle MD 28, Page 169

Centrifugal Coupling 29

Centric Clutch Co., Cranford, N. J.



Style: Centrifugal; standard or spring-controlled shoes

Size: Standard models — thirteen sizes 3½ x 1½ to 24 x 8 in.; max bores 1 to 7 in. respectively, min bores ½ to 3 in. respectively; weight with max bores 7 to 1315 lb respectively

Service: Standard size 3½ x 1½ handles 0.04 hp at 390 rpm, 3.6 hp at 1750 rpm; size 24 x 8 handles 300 hp at 390 rpm, 600 hp at 690 rpm

Design: Outer shoes in driver element and inner shoes in driven unit, engagement results from centrifugal action; outer shoes transmit rated motor hp, inner shoes 40% rated motor torque for 140% normal running torque; above 140% torque, shoes slip; spring control type permits pick-up of load at preset speed

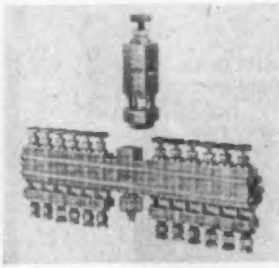
Applications: For coupling normal torque motor drives to coal pulverizers, cloth calenders, extractors, printing presses, grinders, etc. and also for coupling to engine drives.

For more data circle MD 29, Page 169

NEW PARTS AND MATERIALS

Sight-Feed Oilers 30

Oil-Rite Corp., Milwaukee 15, Wis.



Style: SFM, gang-mounted

Size: ¼-in. female pipe thread inlet; ¼-in. tube outlets for compression fittings

Service: Pressure or gravity centralized lubrication with oils; 2 to 24 sight-feed valves

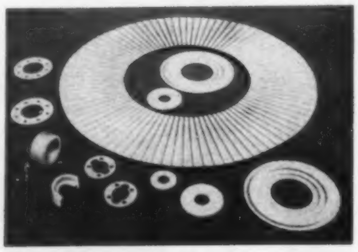
Design: Aluminum alloy oiler body; central mounting shank with standard thread and nut; individual flow adjustment from full flow to shutoff; unbreakable sight

Applications: For centralized control of pressure or gravity oiling systems to individual bearings.

For more data circle MD 30, Page 169

High-Voltage Insulators 32

United States Gasket Co., Camden, N. J.



Form: Sheet, cylinder, rod, tube and bar; molded or machined Teflon

Size: Standard and special; individual insulators as large as 23-in.

Service: High-frequency high-voltage insulation; temperatures from −90 to 500 F; outdoor service

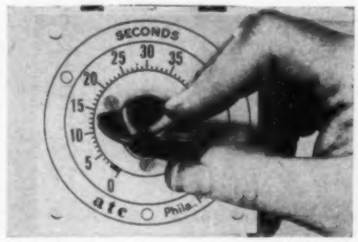
Properties: Chemically inert; power factor 0.0005; dielectric constant all frequencies, 2.0; water abs, 0

Applications: Used as spacers for coaxial cables, inserts for coaxial connectors, beads, etc.

For more data circle MD 32, Page 169

Preset Time Switch 34

Automatic Temperature Control Co. Inc., Philadelphia, Pa.



Style: Type J3, manual reset, NO or NC

Size: 3⅜ in. high, 3⅜ in. wide and 5¼ in. long unenclosed

Service: Standard dial ranges, 0-15 seconds to 0-24 hours

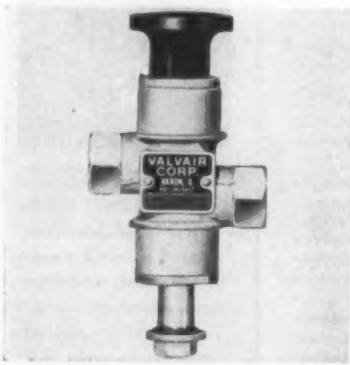
Design: Knob manually turned to desired interval; self-starting Telechron synchronous motor drives time indicator; at 'zero' position, heavy-duty snap-acting electrical control switch of open-blade type trips to shut off timer

Applications: For time control of photographic processes, blueprint machines, motors, etc.

For more data circle MD 34, Page 169

Air Control Valve 31

Valvair Corp., 454 Morgan Ave., Akron 11, O.



Style: Sliding stem; two or three-way; normally open or normally closed; with or without mounting bracket or spring return; knob operated

Size: ¼ and ⅜-in. pipe ports

Service: Air to 175 psi, temperatures to 125 F

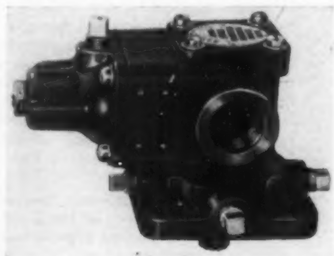
Design: Die-cast zinc alloy body; OU packer seals; cylindrical ends, stem packing spreader, flange end bearings, and springs interchangeable; air sealed in 'on' and 'off' positions without metal seats

Applications: For control in pneumatic circuits on machines.

For more data circle MD 31, Page 169

Fuel Pump 33

Lear Inc., Romec Div., Elyria, O.



Style: Model RG-9080, USAF type G-18 engine driven; positive-displacement vane

Size: 4 23/32-in. high, 2 23/32-in. wide, 3 27/32-in. long including ¾-in. shaft extension; 1.94 lb; ½-in. ANPT ports; mounting centers, 2 x 2 in., 4 holes; shaft spline, 11 teeth, 24/48 pitch, 30° pa

Service: In compliance with Spec MIL-P-5355 for 100 gph; fuels with as high as 30% aromatic blends and low temperature of −65 F; displacement, 0.209 cu in. per revolution; pressure range, 2½ to 20 psi; cw or ccw rotation

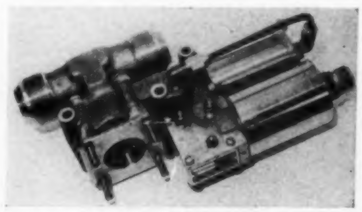
Design: Bypass valve for wobble pump and spring-loaded adjustable relief valve integral; valve housing vent to atmosphere or connects to pressure line from supercharger; pump selfpriming; O-ring seals; shaft seal withstands one-degree misalignment

Applications: For pumping fuel to aircraft engines.

For more data circle MD 33, Page 169

Hot-Air Shutoff Valve 35

Hydro-Aire Inc., Burbank, Calif.



Style: Two-way, electrically controlled, normally open

Size: 3.31 in. wide, 5 11/16 in. high, and 9½ in. long; inlet port flange for 1.500-in. tube fitting, AND-10086; outlet ports 1.250-in. tube, AND-10056

Service: Hot-air; pressures from 35 to 215 psi; ambient temperature range −65 to 700 F; air temperature to 1000 F max; 14 to 30v d-c; max operating current 10 amps instantaneous, max holding current ½-amp

Design: All components including seals stainless steel or carbon; unit opens or closes in ⅛-second, closes if current fails; electric receptacle similar to AN-3102-12S-3P

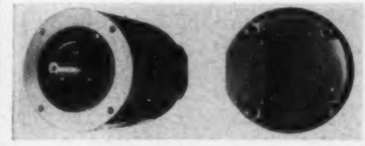
Applications: For shutoff of aircraft hot air circuits such as deicing and other special aircraft or industrial hot-air installations.

For more data circle MD 35, Page 169

NEW PARTS

Small A-C Motor — 36

Electro Machines Inc., Cedarburg, Wis.



Style: General-purpose open, drip-proof (P); general-purpose totally enclosed (PN); and fan duty totally enclosed (PC); face mount

Size: 1/6 to 1 hp; conduit openings with 7/8-in. plain hole or 1/2-in. pipe tap; face mountings NEMA type C, 42, 56 and 66 frames

Service: 3450 to 860 rpm constant speed; three phase a-c; standard voltages 208, 220, 380, 440, and 550; frequencies 60, 50, 40 and 25 cycles; type P, 40C rise; types PN and PC, 55C rise

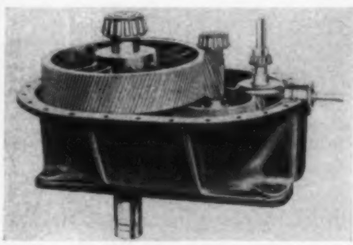
Design: Rolled-steel frame; prelubricated ball bearings; end mounted conduit box

Applications: Constant-speed drive for pumps, coolant pumps, and other face mounted equipment as variable-speed transmissions.

For more data circle MD 36, Page 169

Speed Reducer — 37

The Falk Corp., Milwaukee 8, Wis.



Style: Right-angle, vertical, upward or downward; GDX double reduction, GRX triple reduction

Size: 15 to 1000 hp

Service: Occasional, intermittent or continuous duty; reduction—5.7:1 min., 430:1 max; input—1750 rpm, higher if necessary

Design: Shafts—heat-treated alloy steel in small sizes, SAE 1045 annealed forged steel in large sizes; bearings—straight roller and two tapered roller for high-speed shafts except on larger triple reduction units where straight roller and ball bearings are used, tapered roller on all other shafts; gears—spiral bevel precision cut carburized and lapped in first reduction, helical precision cut and shaved in other reductions; lubrication—forced to top of unit, gravity feed to gears—Falk patented bi-directional oil pump

Applications: General-purpose and special right-angle drives.

For more data circle MD 37, Page 169

Contact KAYDON *of Muskegon*

FOR ALL TYPES OF BALL AND ROLLER BEARINGS 4" BORE TO 120" OUTSIDE DIAMETER

KAYDON Double Row Taper Roller Bearing, 34.000" x 42.250" x 7.375", permits smooth, precision operation of the table spindle on the big 120" capacity double-head, heavy-duty super precision grinder shown at right.

THIS double head grinder permitted production of super precision accuracy in Naval gun mounts. It was designed to produce work of an angular accuracy of less than one-half thousandth (.0005") in 72", in flatness, squareness, concentricity, roundness and taper, and permitted interchangeability of gun mount parts which formerly were tediously hand scraped. Parts produced in regular production to an accuracy of .0002".

We believe this to be the world's largest, super precision industrial grinder, 26 of which were manufactured.

In heavy oil field machinery, ponderous steel-mill equipment, complicated paper-mill units... in rugged road-building machines, excavators, hoists, crushers ... powerful bending machines, production units, and other industrial equipment . . . KAYDON bearings are improving performance and lengthening service-life. Look to KAYDON, Standard or Special, for the better bearing service your equipment deserves.

* * * *

Counsel in confidence with KAYDON. Capacity now available for all sizes and types of KAYDON bearings . . . and for atmospheric controlled heat treating, precision heat treating, salt-bath and sub-zero conditioning and treatment, microscopy, physical testing and metallurgical laboratory services.

KAYDON *Types of Standard or Special Bearings:*
Spherical Roller • Taper Roller
Ball Radial • Ball Thrust
Roller Radial • Roller Thrust

THE KAYDON ENGINEERING CORP.

MUSKEGON • MICHIGAN

HELPFUL LITERATURE

FOR DESIGN EXECUTIVES

80. Air-Hydraulic Cylinders

Logansport Machine Co.—14-page illustrated catalog No. 100, Section 3, presents complete data relative to recent improvements and modifications of Air-Draulic feed controlled cylinders. They combine fast-acting low pressure operation of air with smooth uniform controlled regulation of oil and are suitable for pushing, pulling, pressing, clamping and power movements in any direction.

81. Metallic Bellows

Clifford Mfg. Co.—8-page illustrated bulletin "What to Consider When Selecting a Metallic Bellows" is reprint of technical article by design engineers which deals with all factors related to bellows application. Applications, bellows' construction analyses, materials, flexibility, filling medium and life are discussed.

82. Universal Joints

Curtis Universal Joint Co.—6-page illustrated catalog form C1 tells how to select universal joints; provides graphs on static torque tests, angle of operation, efficiency curves and frictional heat loss curves; presents dynamometer tests; and summarizes engineering data.

83. Blower Wheels

Torrington Mfg. Co.—48-page illustrated catalog "Torrington Airotor Blower Wheels" gives design and application data on all sizes and types of Coasting Hub blower wheels. Included are performance charts, dimensions, power requirements and packing information. Details of complete line of blower wheels are presented.

84. Alloy Castings

American Brake Shoe Co., Electro-Alloys Div.—60-page illustrated manual "High Alloy Castings" outlines engineering, research and control aspects leading to production of Thermalloy and Chemalloy castings which are heat, corrosion and abrasion resistant. Typical castings produced for general industry, properties of both alloys and suggested applications are presented.

85. Hydraulic Pumps & Cylinders

Commercial Shearing & Stamping Co.—8-page illustrated catalog H-1 contains design, specification, performance and application data on line of oil-hydraulic hand-operated pumps and single-acting cylinders. Lever-actuated pumps operate in any position and deliver pressures up to 5000 psi. Standard 10-in. stroke cylinders can be supplied in 1, 1½, 2 and 2½-in. diameters.

86. Aluminum Pig Production

Reynolds Metals Co.—12-page illustrated booklet is descriptive of production of aluminum pig by Hall electrolytic reduction process at Jones Mills, Arkansas, plant. Equipment for producing rated capacity of 144 million pounds of aluminum yearly is pictured and electric reduction process is explained.

87. Propeller Fan Wheels

Robinson Ventilating Co.—16-page illustrated bulletin No. 7100 concerns design No. E-7 propeller fan wheels. Available in steel, alloys, bronze or aluminum with any type mounting, wheels are recommended for use where large quantities of air at pressures under 1½-in. water must be moved.

88. Chrome Plating Unit

Ward Leonard Electric Co.—4-page illustrated bulletin depicts and describes model A-20 Chromaster industrial chrome plating unit which is suitable for precision plating of small machine tool accessories and parts.

89. Small Flexible Shafts

Kupfrian Mfg. Co.—2-page illustrated data sheet 194-649 lists typical properties of light-duty flexible shaft couplings which are applicable for remote control, connecting misaligned shafts and light-duty power drives. Design data, standard end fittings and casing materials are some of the subjects listed.

90. Molded & Laminated Plastics

Richardson Co.—4-page bulletin "Insurok" pictures typical products in extensive line of laminated and molded plastics components. Details are given on available forms, fabricated parts, gear stock, bearings and post-formed laminates.

91. Variable Speed Transmission

Revco Inc.—4-page illustrated bulletin on Zero-Max infinitely variable speed transmission provides information on models with torque ratings up to 120 lb-in. Tabular data is provided to aid selection of proper type torque converter to be used with motors of various speeds and powers.

92. Glass Cylinders

Dunbar Glass Corp.—8-page illustrated catalog No. C-51 lists standardized glass cylinders available in 48-in. maximum length, ½-in. minimum length and 24-in. maximum diameter, 1¼-in. minimum diameter. They can be supplied with bead-edge, end-ground or end-ground and beveled finishes.

93. Oilers & Tube Forms

Eynon-Dakin Co.—16-page illustrated 1951 catalog describes line of oilers, oil caps, vents and special tube forms. Specifications are given on components for use by automotive, electrical and other manufacturing industries.

94. Motors & Generators

Ideal Electric & Mfg. Co.—12-page illustrated bulletin No. 505 covers high speed synchronous motors and generators with speeds of from 500 to 1800 rpm. Generators are supplied for either belted or coupled drive from gasoline, diesel or gas engines, steam turbines and vertical or horizontal waterwheels. Typical motors include those supplied for fans, blowers, motor-generator sets and frequency changers.

95. Tungsten Electrodes

Fansteel Metallurgical Corp.—4-page illustrated bulletin 1.103-1 is entitled "Fansteel Premium Quality Tungsten Electrodes." Usable for inert gas and atomic hydrogen arc welding, electrodes are 99.9 per cent pure tungsten. Available standard diameters and lengths are listed, and seven practical suggestions for obtaining longer electrode life, better welds, labor saving and lower costs are presented.

96. Springs

Accurate Spring Mfg. Co.—40-page illustrated technical guide "Accurate Handbook of Technical Data on Springs" offers helpful shortcuts on making spring calculations. Included is guidance for specifying springs. All factors related to design and application of extension and compression springs are covered.

FOR MORE INFORMATION

on developments in "New Parts" and "Engineering Department" sections—or if "Helpful Literature" is desired—circle corresponding numbers on either card below

FIRST CLASS
PERMIT No. 36
(Sec. 34.9 P.L.&R.)
Cleveland, Ohio

BUSINESS REPLY CARD
No Postage Stamp Necessary if Mailed in the United States

4c POSTAGE WILL BE PAID BY—

MACHINE DESIGN
Penton Building
CLEVELAND 13, OHIO

Readers' Service Dept.

M. D. Numbers

1	21	41	61	81	101
2	22	42	62	82	102
3	23	43	63	83	103
4	24	44	64	84	104
5	25	45	65	85	105
6	26	46	66	86	106
7	27	47	67	87	107
8	28	48	68	88	108
9	29	49	69	89	109
10	30	50	70	90	110
11	31	51	71	91	111
12	32	52	72	92	112
13	33	53	73	93	113
14	34	54	74	94	114
15	35	55	75	95	115
16	36	56	76	96	116
17	37	57	77	97	117
18	38	58	78	98	118
19	39	59	79	99	119
20	40	60	80	100	120

MACHINE DESIGN 1-51
Penton Building
Cleveland 13, Ohio

Please send literature or detailed information on subjects circled at left to—

NAME ____________ TITLE ____________

COMPANY ____________

MACHINES MANUFACTURED ____________

ADDRESS ____________

CITY AND STATE ____________

This card MUST be completely filled out. Please TYPE or PRINT.

97. Hand Welding Torches

Air Reduction Sales Co.—36-page illustrated catalog ADC-702 contains information on line of hand welding and cutting torches, outfits and accessories. Charts show correlation of tip, mixer, extension and other components of various torches for specific jobs.

98. Conveyor Roll Bearings

Marlin-Rockwell Corp.—4-page illustrated form No. 1529 describes M-R-C conveyor roll bearings which have rust resistant Lubri-Seal shield, slotted sides which ease assembly in side frame or mounting bracket, and rust resistant enclosed back.

99. Air Cylinders

Bendix-Westinghouse Automotive Air Brake Co.—4-page illustrated folder 2006 is entitled "Something New in Air Cylinders." Features of diaphragm type cylinders for holding, clamping, bending, staking, riveting and other operations are shown.

100. Plastic Materials

Union Carbide & Carbon Corp., Bakelite Div.—24-page illustrated booklet is entitled "A Simplified Guide to Bakelite and Vinylite Plastics and Resins." Various forms of Bakelite, styrene, polyethylene and Vinylite plastics and resins are classified under 14 major headings. General characteristics, properties and applications of each form are discussed.

101. Gears

Rynel Corp.—4-page illustrated bulletin "Certified Gears" shows typical machines and outlines facilities of company for producing spur, helical, worm, internal, segment, ratchet and straight-tooth bevel gears; worm wheels and sprockets. Gears can be produced with pitch diameters up to 6 in. and with 16 to 700 diametrical pitch.

102. Direct-writing Recorders

Sanborn Co.—18-page folder containing bulletin inserts deals with single and multiple channel direct-writing recorders and instrument amplifiers, Galvanometers, recorder assembly, amplifier, strain gage amplifier, amplifier-recorder and Poly-Viso recording equipment are covered and their prices listed.

103. Centrifugal Pumps

De Laval Steam Turbine Co.—3-page illustrated bulletin No. 1001 contains description, cross-sectional drawing, rating tables and dimensions of type GS centrifugal pump which is built in three sizes from 2½ to 4 in. for capacities to 400 gpm and heads to 230 ft. Features include prelubricated bearings, mechanical seals, stainless steel shaft, horizontally split cast-iron casing and parts interchangeability.

104. Machine Window

Bijur Lubricating Co.—4-page illustrated bulletin 4D shows new and varied applications of one-piece Window Unit which permits indication of liquid level, flow, drip or movements of internal parts. Unit is mounted by simple press fit in machine, sump or equipment and fits flush with surface.

105. Steam Traps

V. D. Anderson Co.—4-page illustrated bulletin No. 550 offers specifications, capacities, sizes, pressures, weights and prices of principal types of Super-Silvertop inverted bucket steam traps. Data are included also on self-cleaning strainers and steel steam traps.

106. Ruling & Drawing Devices

C-Thru Ruler Co.—16-page illustrated general catalog lists various plastic ruling and drawing instruments including rulers, protractors, engineers scales, lettering devices, slide rules, T-squares and computing instruments.

107. Hydraulic Mechanisms

Barry Motors, Inc.—4-page illustrated bulletin No. 5010 covers line of Rotary Power hydraulic pumps, motors and transmissions. Principles of operation, design features and industrial applications are given.

108. Insulating Material

Johns-Manville Sales Co.—4-page illustrated folder "Build and Insulate with One Material" deals with Marinite sheet material composed of asbestos fibers, diatomaceous silica and inorganic binder. It is suitable for use in breechings; for construction of ovens and driers; and for fireproofing of structural steel.

109. Anticorrosion Materials

Cooper Alloy Foundry Co.—4-page folder "Materials Selection Chart" lists carbon steel and Cooper alloys 17S, 17SM and FA-20 in order of increasing cost so that selection can be made readily for any of 350 different corrosive conditions.

110. Coil Springs

Hunter Spring Co.—8-page illustrated bulletin "Spring Buyer's Guide" is designed to aid economical purchasing by reducing to understandable terms the various factors affecting the cost of coil springs. Sections are entitled Engineering, Manufacturing, Materials and Glossary of Spring Maker's Terms.

111. Locknuts

Industrial Fasteners Institute—20-page illustrated application study is entitled "Types of Locknuts and Their Principles of Operation." Section devoted to various types distributed commercially by member concerns of Institute covers principle of operation and description of each style.

112. Titanium Alloy

Kennametal Inc.—4-page illustrated bulletin 1050 is descriptive of Kentanium metal carbide for use where conditions of intermittent or continuous high temperatures in oxidizing atmospheres are combined with abrasion and compressive or tensile loads.

113. Variable Volume Pumps

Denison Engineering Co.—6-page illustrated bulletin P-2 is descriptive of HydrOILic variable volume pumps available in three sizes: 6 and 17-gpm delivery for 5000-psi requirements and 33-gpm delivery for 4000-psi service. Handwheel, servo, pressure compensating and stem type control can be obtained.

114. Screw Products

Mac-it Parts Co.—6-page illustrated pocket-size form 42-A is descriptive of line of heat treated alloy steel screws which includes hollow set, socket head cap, stripper bolt, hexagon socket pipe, square head set and tool post types.

115. Hose Coupler

Snap-Tite, Inc.—8-page illustrated Vol. 1, No. 5 of Snap-Tite Bulletin features 3-in. lightweight coupler for water, gasoline, fuel oil and fire hoses. Made of die-cast aluminum, coupler has positive swivel action, is leakproof and will not corrode. It is designed for applications where high volume low pressure delivery is required.

116. Flexible Shafts

S. S. White Dental Mfg. Co.—32-page booklet 5006 outlines basic principles of flexible shaft selection and applications, recent developments, and construction details and performance data of both power drive and remote control types. Metallic, fabric, rubber and plastic covered casings are covered.

117. Mill & Crane Motor Control

Westinghouse Electric Corp.—4-page illustrated bulletin B-5361 covers line of heavy-duty type M contactors rated from 25 to 2500 amp for direct current mill and crane motor control.

118. Aluminum Bronze Alloys

Ampco Metal, Inc.—4-page illustrated folder "Ampco Alloys and Products" features aluminum bronze alloys which have resistance to wear, abrasion, impact, fatigue, corrosion, erosion, cavitation and sparking. Such products as sheet and plate, pipe and tube, welding electrodes, centrifugal pumps, safety tools and others are covered.

119. Closed Cellular Rubber

Great American Industries, Inc., Rubatex Div.—8-page illustrated catalog No. RBS-12-49 concerns Rubatex expanded compound made from natural or synthetic rubber which incorporates individually sealed cells containing inert nitrogen. It is made in soft, medium and firm forms and in varying densities, sizes and thicknesses. Moisture-resistant material has variety of insulating and shock absorbing applications.

120. Metal Inspection

Dy-Chek Co.—4-page illustrated bulletin 500 and 2-page enclosed periodic bulletin 504 fully describe dye penetrant method of metal inspection and show how this chemical process simplifies nondestructive testing. Dip, brush and spray methods are explained, and varied applications in plant and field are discussed.

M. D. Numbers

1	21	41	61	81	101
2	22	42	62	82	102
3	23	43	63	83	103
4	24	44	64	84	104
5	25	45	65	85	105
6	26	46	66	86	106
7	27	47	67	87	107
8	28	48	68	88	108
9	29	49	69	89	109
10	30	50	70	90	110
11	31	51	71	91	111
12	32	52	72	92	112
13	33	53	73	93	113
14	34	54	74	94	114
15	35	55	75	95	115
16	36	56	76	96	116
17	37	57	77	97	117
18	38	58	78	98	118
19	39	59	79	99	119
20	40	60	80	100	120

MACHINE DESIGN
Penton Building
Cleveland 13, Ohio

1-51

Please send literature or detailed information on subjects circled at left to—

NAME ______________ TITLE ______________

COMPANY ______________

MACHINES MANUFACTURED ______________

ADDRESS ______________

CITY AND STATE ______________

This card MUST be completely filled out. Please TYPE or PRINT.

FIRST CLASS
PERMIT No. 39
(Sec. 34.9 P.L.&R.)
Cleveland, Ohio

BUSINESS REPLY CARD
No Postage Stamp Necessary If Mailed in the United States

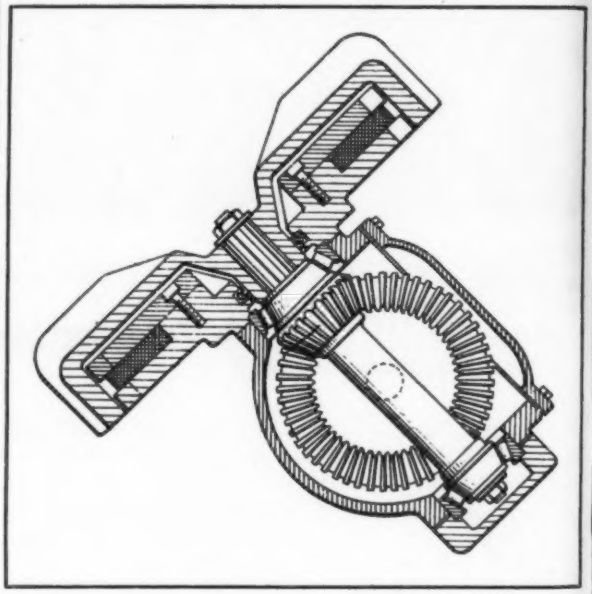
4c POSTAGE WILL BE PAID BY—

MACHINE DESIGN
Penton Building
CLEVELAND 13, OHIO

Readers' Service Dept.

NOTEWORTHY PATENTS

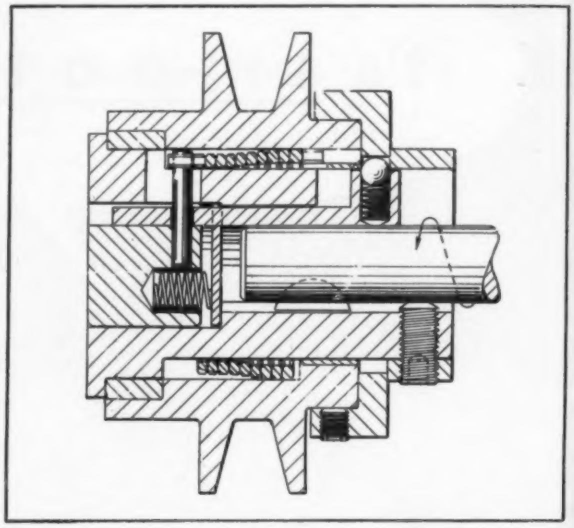
ELECTROMAGNETIC RETARDING of heavy road and rail vehicles is possible with an eddy-current brake that gives a substantial amount of frictionless auxiliary braking at low cost. Patent 2,516,903, assigned to Warner Electric Brake Manufacturing Co. by John G. Oetzel, describes a device using a magnet ring fastened to the rear axle housing and partly surrounded by a metallic drum which rotates with



the rear wheel axles. Since the magnet is stationary, energizing the winding creates a retarding effect on the drum, and consequently the axle, which is proportional to vehicle speed and the energizing current. Because the drum is geared to rotate at from six to ten times axle speed, efficient braking action is obtained even at relatively slow vehicle speeds.

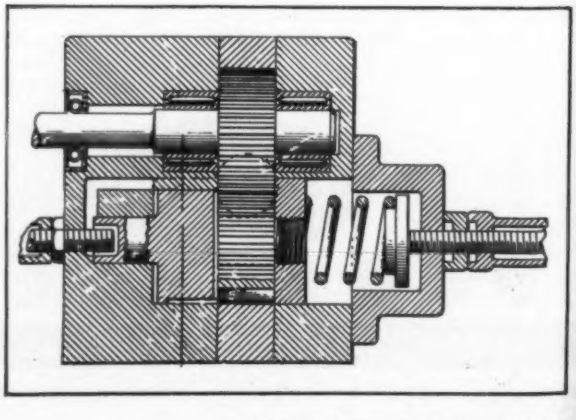
FREE RUNNING of an overload-release spring-clutch mechanism, when disengaged by a predetermined overload, prevents accidental damage to jammed power drives. Pushing an axially spring-loaded control button moves the clutch-spring assembly axially within the pulley and engages a pin on the driving shaft with a hook on the free end of the clutch spring. Since the other end of this spring is connected to the pulley flange, a positive yet cushioned drive results which does not depend on fric-

tional engagement between the spring and shaft or spring and pulley. The clutch is held in the drive position by a spring-loaded ball which drops into a groove in one of the driven members and prevents relative axial movement of pulley and shaft. When torsion in the clutch spring exceeds a predetermined



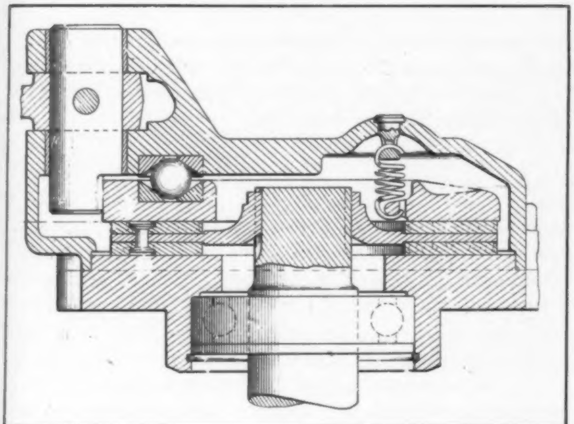
value, the spring allows relative rotation between pulley and drive sleeve, and a cam surface on the pulley flange displaces the lock ball from its groove to disengage the drive. In the disengaged position, the drive shaft pin is completely disconnected from the clutch spring to provide free running between pulley and drive shaft. The patent, No. 2,528,477, has been assigned to The Marquette Metal Products Co. by Weldon E. Rugh.

VARYING DISCHARGE RATE in constant-speed gear pumps is accomplished by sliding the gears axially relative to one another to change the amount of tooth overlap or mesh and thus the fluid discharge rate. A nonrotating block having a notch which matches the gear moves with the sliding gear to seal



off the uncovered portion of the drive gear. Stops are set on the sliding gear to prevent complete disengagement of the gears, but tooth mesh can be reduced to the point where the fluid pumped is just sufficient to make up for internal leakage in the pump, resulting in a zero discharge rate. By using more than one sliding gear in connection with a pressure-responsive control system the pumps, covered in patent 2,526,830 and assigned to H-P-M Development Corp. by Howard M. Purcell, can be made with reversible flow and to provide a constant-pressure output while driven at constant speed in one direction.

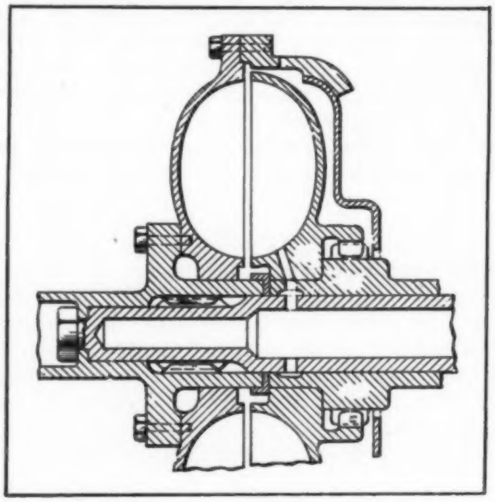
SELF-ENERGIZATION of disk brakes is simply accomplished by using steel balls in cone-shaped recesses to transmit and increase the braking force. Homer T. Lambert has been granted the patent, No. 2,526,143, which has been assigned to Lambert Brake Corp. When the spring-mounted, floating pressure plate is moved radially by a cam, the balls act on the cone-shaped surfaces in the housing and floating disk



to force the friction surfaces together. When contact has been made between the friction disks, tendency of the floating disk to rotate produces additional axial thrust or self-energization. The friction disk rotating with the braked shaft is loosely splined to its shaft to permit the axial movement necessary to engage and disengage. At least two balls, diametrically opposed, are used in conjunction with two or more springs to actuate and retract the floating disk.

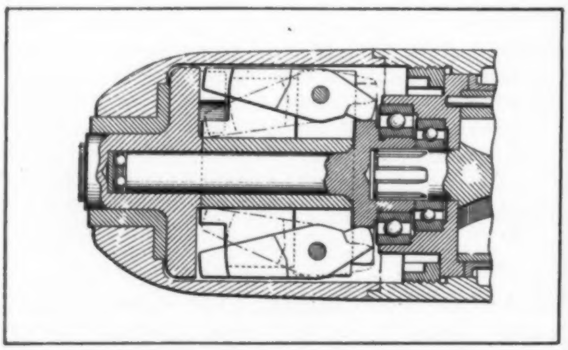
ONE-WAY fluid coupling drive which will not transmit torque in the reverse direction is described in patent 2,521,117 granted to George B. DuBois and Roland B. Ungerer and assigned to Wright Aeronautical Corp. Driving half of the coupling is fastened to a hollow input shaft through which fluid is supplied to the coupling. Driven half of the coupling is splined to the output shaft with a lost motion connection which permits a small relative rotation between these two parts. When torque is in the proper

direction, holes in the flange of the driven half of the coupling line up with holes in the input shaft to admit fluid to the coupling. Reversal of torque rotates



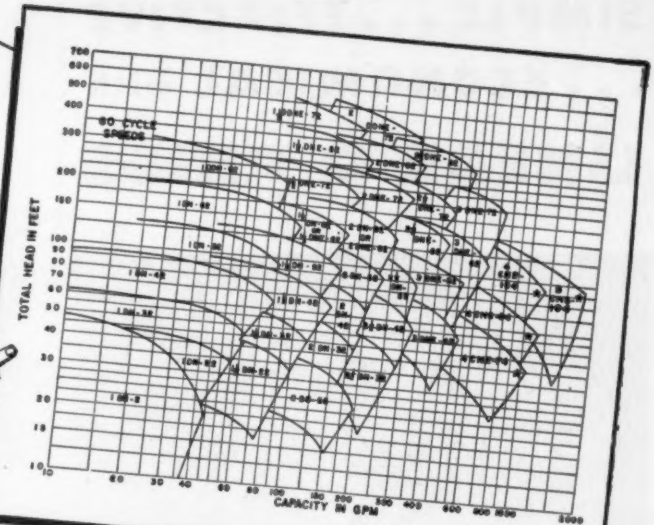
the driven coupling member relative to the input shaft and blocks the supply holes. Fluid in the coupling drains out and prevents transmission of any torque in the reverse direction.

CENTRIFUGALLY - CONTROLLED cam-operated flyweights provide positive drive in an impact clutch mechanism for rotary impact tools. Centrifugal force throws the lower end of pivoted flyweights out to contact jaws on the anvil member. When rotation of the anvil is stopped, cam surfaces on a drive plate splined to the motor shaft engage the upper ends of the flyweights to retract the weights to a smaller radius. Because kinetic energy, which does not change ap-



preciably in this time, is proportional to radius times the speed squared, the drive plate and hammer pick up sufficient rotational speed to again throw the weights out to contact the anvil jaws. Direction of impact can be reversed merely by changing rotation of the motor.

Whatever You Want in a General Utility Pump ...Here It Is!



OVER 10,000 STANDARD COMBINATIONS WITH WORTHINGTON CN AND DN CENTRIFUGALS!

The modern answer to today's small pump problems — Worthington CN and DN Pumps offer a breadth of coverage and a choice of style that will suit your requirements to a T. Hydraulically, both types are identical. Their liquid ends, of the same highly efficient, up-to-the-minute design, are interchangeable, and can be equipped with either conventional packing or mechanical seal.

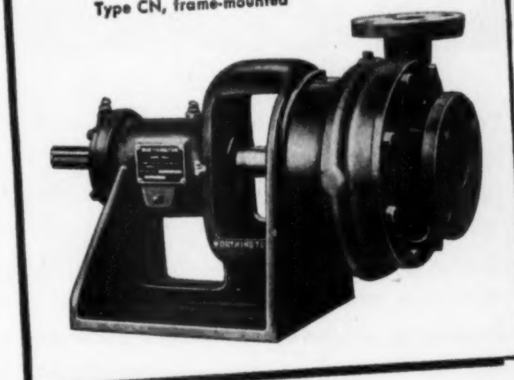
Type CN, frame-mounted, is a real "any-drive" pump — readily driven by an electric motor, V-belt, or any other mechanical power-source. Type DN is the famous Worthington Monobloc design — compact, streamlined, with built-in motor.

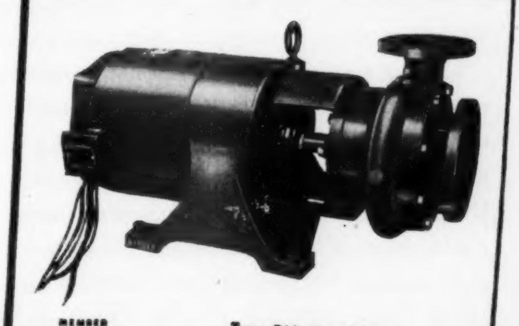
This Is Real Coverage

With the CN and DN you have 40 sizes of pump-casings to choose from, with capacities from 10 to 1750 gpm and heads from 20 to 500 ft., as shown on the above chart. There are 200 combinations of frame-mounted designs and, considering both type pumps, over 10,000 standard combinations! Despite this wide range — the CN and DN pumps are designed to have maximum interchangeability of component parts.

From this vast selection you're sure to find the combination that will meet your pumping needs. That means the end of compromising with pumps that are not just right for *you* — and the beginning of the money-saving performance that proves *there's more worth in Worthington.* For details, call on your Worthington distributor, your "Local Good Right Hand of Industry", or write to *Worthington Pump and Machinery Corporation, Pump and Compressor Merchandising Division, Harrison, New Jersey.*

Type CN, frame-mounted





Type DN, Monobloc

WORTHINGTON

WORTHINGTON PUMP AND MACHINERY CORPORATION
PUMP AND COMPRESSOR MERCHANDISING DIVISION
HARRISON, NEW JERSEY

The Good Right Hand of Industry

POWER TRANSMISSION:
sheaves, V-belts, variable speed drives
PUMPS:
centrifugal, power, rotary, steam
AIR COMPRESSORS:
water-cooled, air-cooled

PC-94

on more than 1500 subjects.

Adequately cross-referenced to guide the reader the manual parallels a dictionary in layout and in use. Many basic terms are defined in a line or two. Topics, broader in scope, such as metals vs plastics, polyester resins and compression molding presses, are treated on several pages or more.

An important feature for the designer is the coverage with respect to fields of application, such as aircraft, automotive, electrical appliance, etc. These major fields of plastics applications and others are discussed in some detail, each entry giving the components for which plastics are used, the specific plastic used and the properties which make that plastic suitable.

The appendix, which is equally valuable, is divided into three sections: a list of 1400 tradenamed plastics, including foreign makes, with composition and manufacturer identified; fabrication charts covering tooling and operating conditions for the various plastics; and bar charts showing comparative costs and properties of various plastics.

Government Publications

Heat Treatment and Properties of Iron and Steel—NBS Circular 495: This 33-page, 8½ by 11-inch booklet presents basic theoretical and practical principles involved in the heat treatment of ferrous metals. Topics discussed include properties of iron, alloys of iron and carbon, decomposition of austenite, heat treatment of steels and cast iron, hardenability, nomenclature of steels, and properties and uses of steels. A list of selected references is also given, and a liberal number of graphs, tables and photographs illustrate the text. Prepared by Samuel J. Rosenberg and Thomas G. Digges, the circular is available from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C., for $0.25 each.

*Formulas and Tables of Coefficients for Numerical Differentiation with Function Values Given at Unequally Spaced Points and Application to Solution of Partial Differential Equations—NACA TN*2214: General differentiation formulas for successive derivatives of a function are obtained in terms of the values of the function at unequally spaced arguments and the corresponding distances between the successive arguments using Lagrangian polynomials of various degrees. The remainder term is also obtained. Tables of coefficients in the formulas for the first four derivatives are given in intervals of 0.01 for the special case where only one spacing at either end is different from the others, as is often encountered near a curved boundry, for different ratios of this spacing to the others.

A general discussion of applying these formulas to the numerical solution of partial differential equations is made. In particular, the application to the equations of the elliptic type is illustrated with a problem involving derivatives of both first and second orders and with the value of the function given on a

circular boundary. After replacement of the derivatives by the formulas based on fourth-degree polynomials, the resulting set of equations is solved by both relaxation and matrix methods. Also indicated are similar applications to problems of compressible flow past isolated and cascade airfoils and through turbomachines, and temperature distribution and thermal stresses in cooled turbine blades. Copies of this report may be obtained from the National Advisory Committee for Aeronautics, 1724 F St., NW, Washington 25, D. C.

Manufacturer and Association Publications

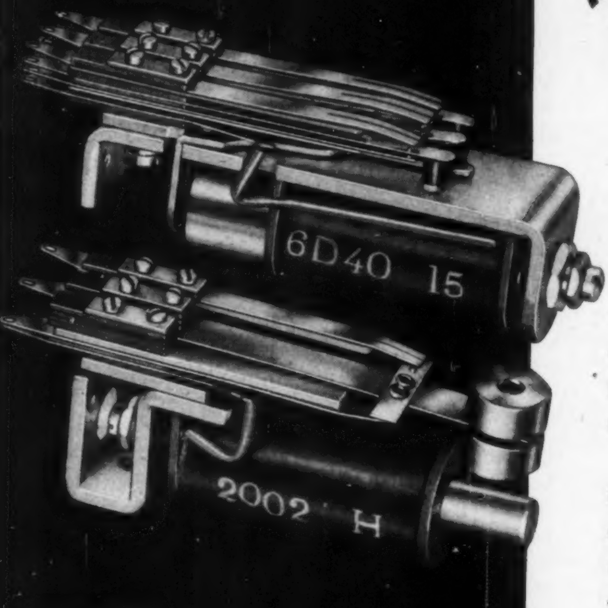
Drafting Standards for Aluminum Extruded Products: Purpose of this manual is to outline certain basic drafting practices for extruded products and to standardize these practices as far as practical at this time. Aluminum extruded products are first defined and identified. Following sections cover standard tolerances, proper dimensioning and standard abbreviations. Recommendations on arrangement and lettering of drawings are also given. The text is sufficiently supported by illustrations and tables. Single copies of this 52-page 6 by 9-inch booklet may be obtained by directing company-letterhead requests to The Aluminum Association, 420 Lexington Ave., New York 17, N. Y.

The ABC's of Welding High Tensile Steels: In simple question and answer form, this booklet compares mild steel and low hydrogen electrodes as related to preheating, underbead cracking, moisture in the arc and in the coating, burn-off rate, cost of operation, applications, and stress relieving of the weldments. Copies of the 7-page booklet may be obtained by writing to the Arcos Corp., 1500 South 50th St., Philadelphia 43, Pa.

Bibliography on Nonmetallic Bearings: This is an annotated bibliography of 101 selected references to the literature of the past twelve years, covering nonmetallic bearings in all such aspects as their manufacture, design, properties, wear, lubrication, performance, testing and application. Some bearing materials are rubber, wood, laminated phenolic plastics, resin-impregnated cotton fabric, micarta, and nylon. Applications discussed are for rolling mills, marine propeller shafts and rubber posts, automatic presses, etc. Copies of the 13-page paper-bound mimeographed booklet may be obtained from the Engineering Societies Library, 29 West 39th St., New York 18, N. Y., for $2.00 each.

Manual of Welding Engineering and Design: A 43-page, $5\frac{1}{2}$ by $8\frac{1}{2}$-in. booklet containing technical data on characteristics, applications and operational procedures of Eutectic alloys and fluxes. All phases of the art of welding are also discussed. This first manual of the Eutectic National Defence Series may be obtained by writing to the Technical Information Service, Eutectic Welding Alloys Corp., 40 Worth St., New York 13, N. Y.

NORTH'S



"Wigglestick"

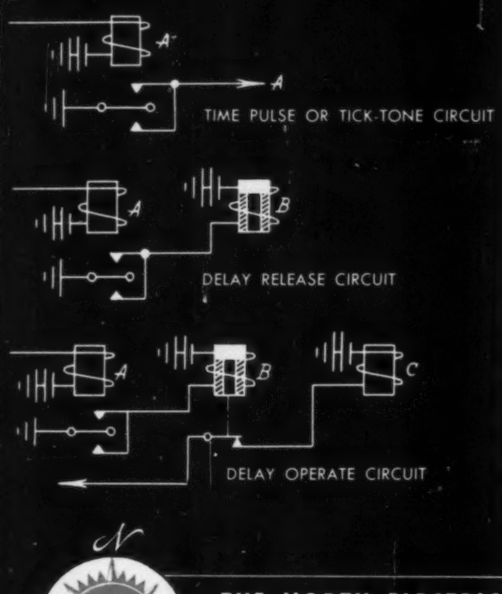
IS A VERSATILE

RELAY

The odd little relay at left with the protruding schnozzle was nicknamed "Wigglestick" by an engineer in one of his lighter moments. Its usefulness to designers of electrical circuits is not to be regarded lightly, however.

Used alone, this vibrating reed relay in a self-interrupting circuit will generate time pulses. Hooked up with a slow-release relay, like the one shown above it, the pair may be used in special delayed circuits, adjustable to any value from one to eight seconds. Other combinations provide a still wider range of application.

These stock relays are typical of a wide variety made by North, one of the oldest manufacturers of relays and switching equipment in the country. The choice ranges from midgets to multiples, from single-makes to pile-ups controlling up to thirty-two circuits with one double-wire control circuit.



NORTH RELAYS

Set a high standard of performance and permanence because this company specializes in all-relay switching equipment. North relay engineering service is second to none.

SPECIAL RELAYS? Let North Engineer Them For You

THE NORTH ELECTRIC MANUFACTURING CO., Galion, Ohio, U. S. A.

Originators of **ALL RELAY** *Systems of Automatic Switching*

FOR PROMPT REPLY
ADDRESS DEPT. D

Shock Resistance

(*Continued from Page* 136)

compressed when the pushbutton is in its normally open position. If this initial compression is made large, the device becomes capable of withstanding considerable acceleration without the pushbutton experiencing relative motion.

3. Distance between contacts should be made as great as convenient operation of the device will permit. In general, the magnitude of the shock which can be successfully withstood increases as the distance between contacts increases.

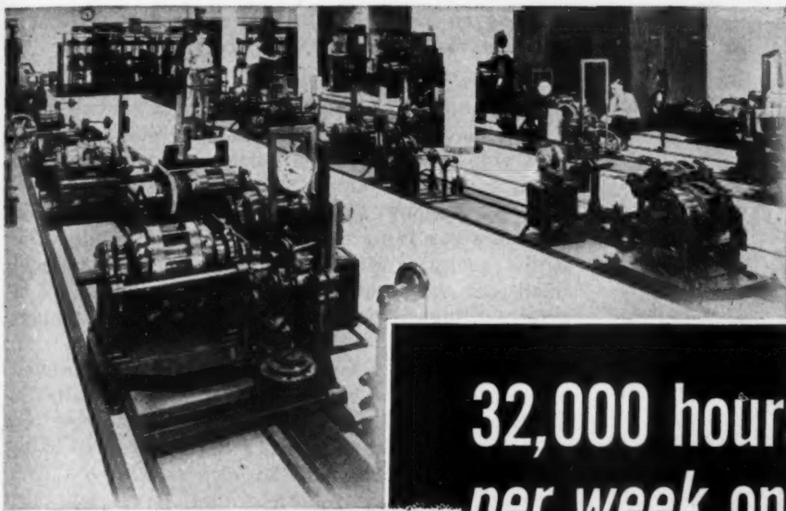
Beneficial results attainable by increasing the stiffness of functional members is well illustrated by a redesign of the bimetallic strip commonly used as a heater element in overload circuits. The conventional strip, shown schematically in *Fig. 25a*, comprises two strips of dissimilar metals welded or soldered together flatwise. In order to obtain the necessary flexure, such a strip is required to be relatively flexible. It is therefore easily deflected by inertia forces, and is not well adapted for use in equipment which is required to be shock resistant.

Rigidity in Bimetallic Mechanisms

A method of introducing rigidity into a bimetallic mechanism is illustrated in *Fig. 25b*. The two temperature-responsive strips actuate the mechanism by means of axial extension and contraction rather than flexure. When subjected to temperature changes as a result of conducting an electric current, one of the strips elongates more than the other and the lever is rotated in a plane parallel with the plane of the paper. The right end of the lever engages electrical contacts, which are embodied in the control circuit of the equipment. Important features to note in connection with the redesigned mechanism are:

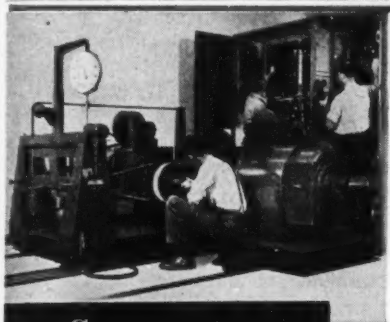
1. Temperature-responsive members are not required to flex. They can be built with great rigidity since their deflection, in the axial direction, is not impeded by a large cross-sectional area. The mechanism therefore can be made sufficiently rigid to insure that the contacts are not accidentally engaged as a result of shock.
2. Magnification of motion, which is necessary in a mechanism of this type in order to obtain adequate travel at the contacts, is obtained from the pivoted lever in contrast to the flexing strip. The lever may be made very rigid, which contributes to the ability of the mechanism to withstand shock.

Irreversible Mechanisms: Shock resistance often can be attained by employing an irreversible component or combination of elements. An irreversible component is defined as one which responds only to a force or torque applied at the input side, remaining unresponsive to a force or torque applied at the output side. A common irreversible device is a screw-and-nut, in which the nut experiences translatory motion as a result of a torque applied to the screw. The screw cannot be made to rotate, however, by



In the GATES Research Laboratories...

32,000 hours of testing *per week* on V-Belts alone!



In Gates V-Belt testing laboratories, which are by far the largest in the world, we run *32,000 hours of testing per week* on V-Belts alone. Thats' the equivalent of *5 years of continuous service* for one belt—*all in a single week!*

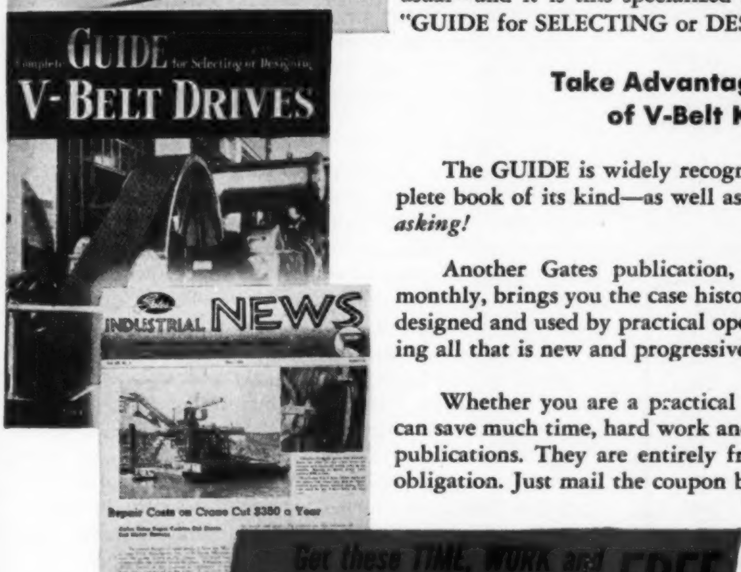
This intensely *Specialized* Research on V-Belts has naturally given Gates Engineers a knowledge of V-Belts and V-Belt drive design that is truly unusual—and it is this specialized knowledge that is embodied in the Gates "GUIDE for SELECTING or DESIGNING V-BELT DRIVES."

Take Advantage of this Great Store of V-Belt Knowledge—FREE!

The GUIDE is widely recognized as the most comprehensive and complete book of its kind—as well as the *easiest to use*—and it is *yours for the asking!*

Another Gates publication, the "INDUSTRIAL NEWS," published monthly, brings you the case histories of actual V-Belt Drives that have been designed and used by practical operating men or by plant engineers—reporting all that is new and progressive in V-Belt drive practice.

Whether you are a practical operating man or a design engineer, you can save much time, hard work and money by having these very useful Gates publications. They are entirely free—are sent to you without the slightest obligation. Just mail the coupon below—today!



Gates VULCO ROPE DRIVES

Engineering Offices and Jobber Stocks IN ALL INDUSTRIAL CENTERS of the U. S. and 71 Foreign Countries

THE GATES RUBBER COMPANY

The World's Largest Makers of V-Belts

DENVER, U. S. A.

The Gates Rubber Company
941 So. Broadway, Denver 17, Colorado

I would like you to send me without cost or obligation—

☐ The Gates "GUIDE for SELECTING or DESIGNING V-BELT DRIVES".

☐ Gates "INDUSTRIAL NEWS" published monthly.

Name________________

Position Held________________

Name of firm________________

City________________ State________

ENG-507

applying an axial force to the nut unless the helix angle is made substantially larger than normally used.

Billiard-Ball Action: A movable member of rigid construction, which normally rests against a rigid stop, may acquire a velocity as a result of a stress wave in the stop. This has been termed "billiard-ball action" because of its similarity to the principles governing the motion of billiard balls. The strain associated with this stress wave is extremely small, but the great stiffness of the balls results in a force being applied to the balls of sufficient magnitude to cause an appreciable velocity.

If a spring of relatively low stiffness is inserted between one of the balls and the stop, the small displacement resulting from the stress wave traveling through the stop then causes a negligible force to be applied to the balls, and no motion results. This principle should be followed where two rigid but relatively movable members rest in contact.

Strain Gages

(*Concluded from Page* 148)

22 simultaneous records giving a series of data showing how the suspension system responded to radical changes. These data gave a good idea of the general character of truck action and indicated that the motion was initiated by the axle. This suggested two ways to proceed in an attempt to correct the condition; one was to control lateral motion of the axle and the other to control angular motion of the truck frame.

An attempt to control lateral motion of the axle is illustrated by the data of *Fig.* 5. On examination of the various curves it will be noticed that various degrees of hydraulic damping produced little or no effect on car body acceleration trace *a* which after all is the pay-off, but an important change was produced in the action of the truck frame. Traces *c* and *d* if carefully examined, show a 90-degree shift in phase. Record I shows that the two traces are about 90 degrees out of phase while in IV and V they are 180 degrees out of phase. This last bit of data provided the key to the ultimate solution as it indicated where control would be the most effective. The next step, therefore, was application of an angular damper between the truck frame and car body with the results that can be seen in *Fig.* 8. The accomplishment needs no elaboration.

This is but one example that illustrates the usefulness of this method of measurement. It is easy to visualize any number of instances in design where similar knowledge of the complex interrelation of all the variables involved would make it relatively easy to single out the element that caused all of the trouble.

"Do not go through your life collecting data from handbooks or tests and call yourself an engineer. Regardless of the excellence of the reports you write, leadership is acquired only by the successful taking of risks"—Crosby Field, *president, Flakice Corp.*

End-Loaded Beams

(Concluded from Page 140)

$(3000/0.109) + (475\times0.5/0.01268) = 27{,}500 + 18{,}700 = 46{,}200$ psi.

EXAMPLE 2: *Fig.* 9 shows a simple supported beam of 1-inch square cross-section carrying a concentrated load of 1250 pounds at its center. For added strength, the beam is braced by the cable and central strut shown, the strut being screwed up until the force in it is 1000 pounds. Plot the bending moment diagram for the beam and calculate the magnitude of the maximum resultant stress. Assume $E = 30\times10^6$ psi.

The length of the cable is $2\sqrt{(12.5)^2 + (30)^2} = 65$ in. By resolution of forces at the lower end of the strut, the tensile force in the cable is $(1000\times32.5)/(12.5\times2) = 1300$ pounds. The beam then carries an axial compressive load of $1300\times30/32.5 = 1200$ pounds, together with a resultant lateral load of $(1250-1000) = 250$ pounds. The relevant moment of inertia for bending of the beam is 1/12 in.[4] and hence

$$\frac{kl}{2} = 30\sqrt{\frac{1200\times12}{30\times10^6}}$$

$$= 0.657 \text{ radians}$$

$$= 38 \text{ degrees}$$

$$\frac{W}{k} = \frac{250\times30}{0.657} = 11{,}400 \text{ lb-in.}$$

The bending moment diagram is then drawn as shown in *Fig.* 10, from which it may be seen that the maximum bending moment of 4400 lb-in. occurs at the center of the beam. The maximum resultant stress due to direct compression and bending is then $1200/1\times1 + (4400\times12\times0.5) = 1200 + 26{,}400 = 27{,}600$ psi.

REFERENCES

1. H. B. Howard—"Aeronautical Research Committee Report and Memoranda No. 1233".
2. H. B. Howard—*The Stresses in Aeroplane Structures*, Sir Isaac Pitman & Sons Ltd., London, England, 1933.
3. H. D. Conway—*Aircraft Strength of Materials*, Chapman & Hall Ltd., London, England, 1947.

They Say...

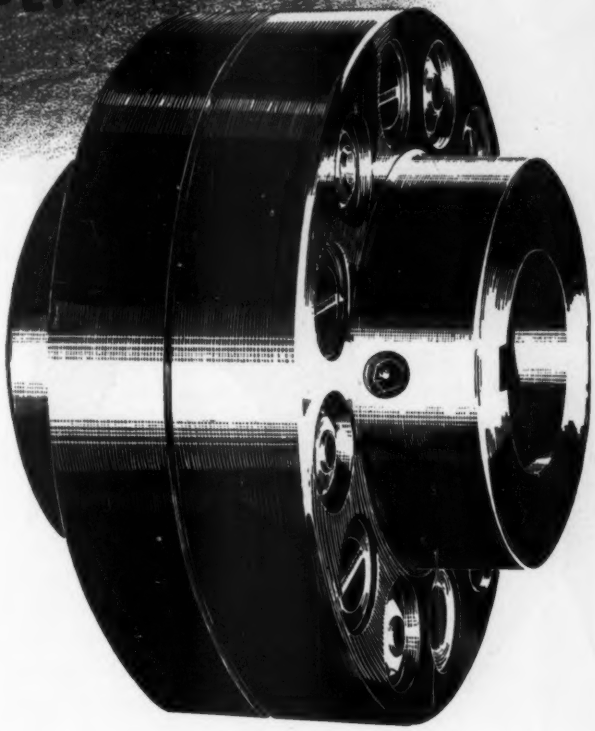
"Lack of product design, of machine tools, and of materials are definite handicaps to the producibility of jet engines. A manufacturer cannot produce these powerplants unless he has buildings, design, machinery, materials, money, people, and tools, and the lack of any of them would delay production. Once annually the military should address to each manufacturer a simple question seeking to ascertain what would prevent his company from delivering a specified type of engine in specified quantity within 18 months. Manufacturers would report the bottlenecks they can see no way to circumvent, and manufacturers and the military then would join in developing a program which would make the required production possible"—E. B NEWILL, *Allison Division, GMC.*

WHEN DESIGN ENGINEERS THINK OF RUBBER BUSHED FLEXIBLE COUPLINGS

THEY THINK OF

AJAX

as America's standard



• 30 Years of satisfactory performance on direct-connected generators, pumps, compressors, winches, speed reducers and other similar equipment have proved the economy of protecting good machinery with Ajax Flexible Couplings.

They protect bearings, gears, armatures, impellers and other costly working parts against unavoidable angular and offset misalignment.

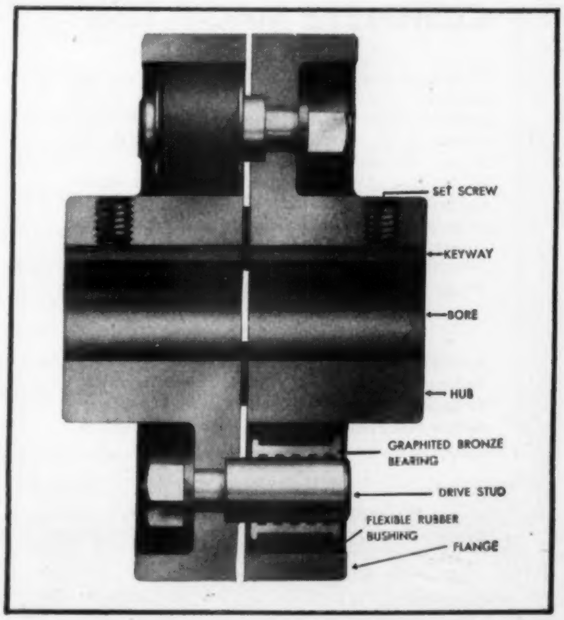
Ajax Flexible Rubber Bushings provide a quiet, cushioned drive in both directions of rotation.

Ajax oilless bronze bearings are self-lubricated for life.

No problems are set up by centrifugal force. They operate successfully in abrasive-laden air. They operate vertically or horizontally.

Free end float lets electrical machinery find its magnetic center without interference.

Standard Ajax Couplings are made of forged steel or cast semi-steel. Aluminum, bronze and other metals available for special applications. Made in a wide range of sizes and capacities...write for catalog.



AJAX FLEXIBLE COUPLING CO. INC.

WESTFIELD, NEW YORK

Design Abstracts

(Continued from Page 154)

operation if any degree of snug fit is to be expected. Also, the minimum hole diameter varies with the temperature range of the alloy being cast. Low-temperature, nonferrous alloys can contain holes as small as 0.010 inch. The minimum for high-melting metals is usually 0.015 inch. In all cases the depth of hole should not exceed 5 times the diameter for holes under 0.125 inch. From 0.125 to 0.250 inch the depth may be as much as 10 times the diameter, under special conditions where reinforced insert cores may be employed. Where the holes can be used as-cast or with a reaming cut, they are an economy in production, particularly with numerous small-diameter holes that would otherwise necessitate involved jigging and drilling operations.

Minimum wall thicknesses are also governed by the temperature of the metal being cast, as well as other design factors. For limited areas in low-temperature alloys, walls can be held to 0.025 inch within a few thousandths. In high-temperature metals, walls less than 0.040 inch are usually impractical for any other than very short distances. This is also true of projections from the casting body. But where a uniform taper is part of the casting design, as in turbine blades, edges as thin as 0.010 inch with final radii as slight as 0.005 inch may be produced reliably in all metals.

Flat Surfaces Obtained by Grinding

The flatness of a cast surface varies inversely with its section thickness, because of the cavitation at the center caused by metal shrinkage. Although this can be overcome by heavy gates and risers to each such section, the added foundry costs involved in the attachment, removal, and remelting of such attachments are considerable. It is usually cheaper to grind off enough metal to obtain a flat surface, wherever this is essential to the operation of the casting.

The investment-casting process is unique in being able to produce commercially valuable shapes from any metal which can be liquefied commercially and cast. In the broad sense, the design engineer is freed from the necessity of compromising his performance and optimum shape for reasons of inability to secure the proper type of metal from the investment mold. However, this freedom of alloy choice should be tempered by considerations of the actual performance characteristics of the cast form of various families of alloys. Particularly, foundry characteristics as they affect ease of production and uniformity of the end product should be considered. In many cases the cheapest metal, which may be quite adequate to perform the intended job, may not produce the cheapest casting, due to its response to the techniques of melting and casting practices by the investment foundry. Inasmuch as the raw metal used for an investment casting may represent less than 1

per cent of the final value of the finished casting, whereas the foundry processing may represent more than 80 per cent of the final price, it is economic to consider base metals of higher initial cost when their castability is better than the lower-cost material. The inverse of this may also be true, where a less costly alloy may perform quite adequately for a specified job. The specifications should stress performance requirements rather than precise chemical composition of the casting.

Attainment of maximum economy in procurement and utilization of investment castings will be aided by the specification of performance rather than chemical standards. While it is true that almost any alloy can be investment-cast, the arbitrary specification of alloys which differ not at all in application and only slightly in basic chemistry from those which are in large-scale production will make the castings more costly because of the more complicated foundry control.

In the lower-temperature casting alloys a part of simple design within the tolerances common to both pressure die casting and investment casting can generally be most economically produced by die casting when quantities exceed 25,000 to 50,000 units. But where the design necessitates complicated cores, it is far simpler to build and operate investment-pattern dies operating at room temperature than pressure dies operating at 800-1000 F. Complex parts of involved internal structure, almost independent of quantity, can usually be most economically cast in investment molds when within the proper range of size and metal volume.

There is no other process which combines so well the factors of dimensional control, reproducibility of design detail, and reliability of structure in small, complex metal objects. In the field of unmachinable alloys, the investment process alone is capable of producing detailed parts to any reasonable degree of dimensional accuracy.

Process Aids Product Development

As an aid in experimental design and development work, the investment process provides a unique method to make rapid changes in alloy, shape, and processing. By use of temporary dies at low cost, enough parts can be produced for the development stage. The tested design can then justify high investment in final production tools, whether for forging or die casting. The product itself is also insured against unexpected design flaws, because the investment castings closely duplicate the physical properties which will be expected of the production parts.

At one and the same time the "lost-wax" casting process is the newest and one of the oldest methods of precision liquid-metal forming; no one can yet estimate the ultimate utility which will result from it. It is an effective tool in overcoming design difficulties when handled intelligently by informed engineers.

From a paper entitled "The Fields of Utility of Investment Castings," presented at the ASME Annual Meeting in New York, November 26-December 1, 1950.

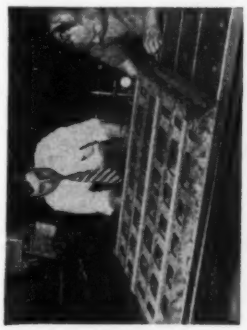
THE ACTUAL IS LIMITED: THE POSSIBLE IS IMMENSE

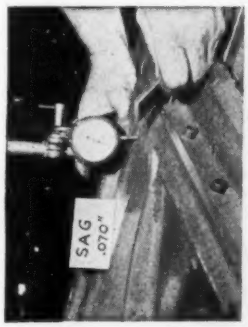
NEW LINCOLN PLANT CREATED BY INCENTIVE-INSPIRED CO-ACTION IN DEVELOPING POSSIBILITIES IN PRODUCT

MACHINE BUILDER CUTS DEFLECTION 80%

BASIC objectives of increased rigidity and greater machine accuracy are today being achieved at lower cost with welded steel designs. As illustrated in the redesign of the Facing Planer for the Porter Machinery Company, the present welded steel table when supported on three corners indicates a deflection of only .014″ where as the former cast construction sagged as much as .070″. Simultaneously, a reduction in weight from 410 lbs. to 320 lbs., made possible by the increased stiffness of welded steel, enables a tolerance of .001″ to be maintained on surface flatness.

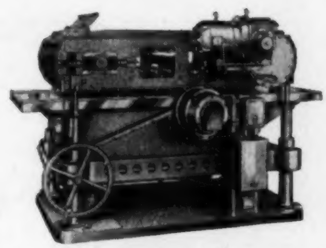
How to Steel Design With Welding is presented in the new 9th Edition "Procedure Handbook of Arc Welding Design and Practice." Contains latest data on machine design together with cost figures. Price only $2.00 postpaid in U.S.A.; $2.50 elsewhere.





Left: Checking table deflection at Porter Machinery Company, Grand Rapids, Michigan. Center: Unsupported corner of former 36″ planer top sags .070″. Right: Present welded steel table deflects only .014″.

the ACTUAL



Former Design. *Had heavy base serving no other purpose than to unify planer.*

increasing the YIELD

Present Steel Design *has rigid box-type base . . . eliminates costly weight.*

the IMMENSITY of the POSSIBLE

HIGHER MACHINE ACCURACY

SEE HOW WELDED STEEL CUTS MATERIAL NEEDED

Machine Design Sheets free on request to designers and engineers. Write Dept. 11,

THE LINCOLN ELECTRIC COMPANY

CLEVELAND 1, OHIO

NEWS
OF MANUFACTURERS

Air Reduction Co. Inc. has started construction of a 272,000-sq ft plant on a 25-acre plot at Union, N. J. The new plant will be used by the Airco Equipment Mfg. Div. of Air Reduction for the manufacture of welding and cutting torches, tips, regulators, oxyacetylene cutting machines, inert-gas arc-welding machines, etc. It is expected that the new plant will be occupied in early spring.

♦

Aluminum Co. of America is launching two major programs to boost America's production of defense aluminum by more than 25 per cent annually. The first, a "quick-action" plan, is underway and shortly will result in the production of added supplies of aluminum at the rate of 158,000,000 pounds per year. This capacity will be reached by using company-owned stand-by facilities which require the use of higher-cost electric power than is economical for peacetime smelting. The second phase is an expansion that involves the erection of new permanent facilities capable of producing 240,000,000 pounds per year. Smelting facilities at ALCOA's Point Comfort, Tex., plant will be enlarged and a new plant utilizing electric power generated from gas or coal fuels will be constructed.

♦

Brush Development Co., Cleveland, O., manufacturer of surface measuring equipment, has expanded its line by purchasing the business of the Faxfilm Co., also of Cleveland. Faxfilm will now be located at the Brush plant.

♦

Borg-Warner Corp., Chicago, Ill., has separated its Ingersoll Steel Division into two distinct and independently operated manufacturing units. One unit, which will continue to be known as the Ingersoll Steel Division, will operate steel mills at New Castle, Ind., and will produce specialty alloy steels for automotive use and for farm implement disks, saws and shovels, soft-center plow steels, high-speed hack saw steels, and solid and clad stainless sheets. The other division, with plants in Chicago and Kalamazoo, will be known as the

If you use machined parts in your products and the present supply situation is giving you a headache . . . *it's time you switched to Gramix,* the powdered metal parts that can be produced in quantity for reasonably prompt delivery. Gramix iron parts of intricate shape can be die-pressed to tolerances within .0005", and cut costly machining time. Gramix parts are oil-impregnated for self-lubrication thus eliminating the added expense of extra oil supply and servicing. Gramix parts are tough, strong, and dependable under all operating conditions. Gramix can help you *improve the mechanical performance of your products,* cut manufacturing costs, and eliminate production snarls due to uncertain parts supply. Go to Gramix . . . the leader in the powdered metal field. Write to us here in Saginaw today for the full facts.

82

THE UNITED STATES GRAPHITE COMPANY

DIVISION OF THE WICKES CORPORATION • SAGINAW, MICHIGAN

Ingersoll Products Division. Products produced by this division include truck wheel disks, washing machine tubs, steel sinks, heat-deflecting screen, etc. at the Chicago plant and automobile stampings and forgings, agricultural implement parts, furnaces, etc. at the Kalamazoo plant.

♦

Bendix Aviation Corp. has purchased the property and facilities of the Victor Animatograph Corp., Davenport, Iowa, to handle increased production of aircraft instruments and accessories for the expanding military program. The plant, built in 1947 and containing 154,000 sq ft of manufacturing space, will become a new division of Bendix. Victor, a subsidiary of Curtiss-Wright Corp., used the plant for the manufacture of motion picture projectors and replacement parts; these will continue to be available from Victor.

♦

Jas. P. Marsh Corp., which recently occupied its new plant in Skokie, Ill., is now expanding that plant to increase production of Marsh pressure gages, dial thermometers, radiator valves, steam traps, and other heating specialties. Need for the addition, which measures about 225 by 75 ft, was further emphasized by Marsh's recent acquisition and development of the Electrimatic line of refrigeration controls and solenoid valves as companions to Marsh's gages and testing equipment for the refrigeration field.

♦

Tri-Engineering Corp. is to be supplied with capital for expansion by Marshall Morrison, prominent Beverly Hills designer and builder who has amalgamated with the corporation. Existing buildings will be enlarged and additional equipment will be purchased to provide for increased production of precision aircraft hydraulic assemblies for the Army and Navy.

♦

B. F. Goodrich Chemical Co., Cleveland, O., has reactivated the government's synthetic rubber plant at Institute, W. Va., to the "check, clean and recondition" stage. In stand-by status since 1947, the Institute GR-S plant is undergoing a complete revitalizing and modernizing so that various process improvements made in the production of GR-S rubber during the last three years can be incorporated. Blaw-Knox Construction Co. crews are getting the 90,000-ton-per-year plant in running order. B. F. Goodrich has operated a GR-S plant at Port Neches, Tex., for

the government since 1943. That plant has a capacity of 60,000 long tons per year.

♦

Radio Receptor Co. Inc., Brooklyn, N. Y., manufacturer of radio and electronic components and complete assemblies, has purchased a 90,000-sq ft factory structure with railroad siding and inside truck-loading platforms at Wythe Ave. and North Third St., Brooklyn. The four-story concrete building will provide additional space for the company's expanding divisions.

♦

Westinghouse Electric Corp. has opened negotiations for a 65 to 100-acre tract of land near the Friendship Airport, Baltimore, Md., for a new plant to meet expanding military demand for products from the company's Electronics and X-Ray Division. The proposed plant will have a manufacturing area of about 400,000 sq ft.

♦

Eutectic Welding Alloys Corp., N. Y., has begun construction of a new Engineering Services Building at the site of its No. 2 plant in Flushing, Long Island, N. Y. Scheduled for completion early in 1951, the building will furnish additional space for Eutectic's long-range welding development research program, as well as house engineering, technical and administrative personnel. The building will also include conference rooms for the convenience of customer representatives.

♦

Hobart Brothers Co. has started operation of new facilities at Troy, O., to test its products under subzero conditions. In a special room, with an intermediate chamber to reduce cold losses resulting from entering or leaving the cold chamber, Hobart products are tested at temperatures down to 65 degrees below zero.

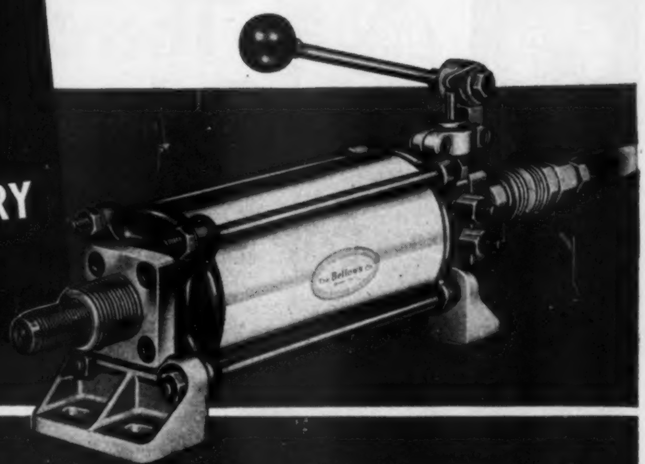
♦

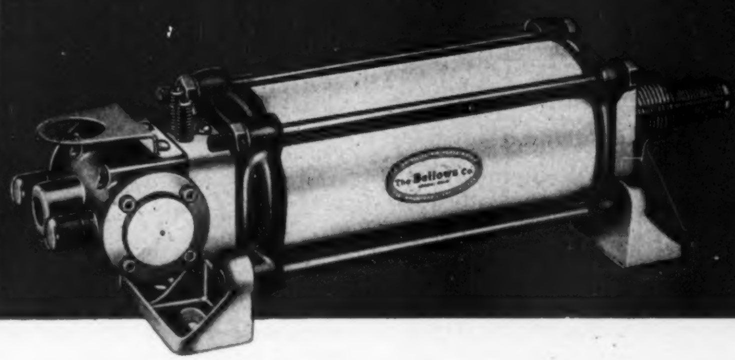
Barry Corp., Cambridge, Mass., is expanding its shock-mount and vibration isolator production facilities. The new plant, now under construction in Watertown, Mass., will more than double Barry's present floor space.

♦

Standard Coil Products Inc., manufacturer of small coils and other devices for electronic equipment, has arranged to acquire the Kollsman Division, New York city, of the Square D Co. Aircraft instruments

IF YOU DESIGN FOR AIR OPERATION YOU'LL WANT TO KNOW ABOUT THESE REVOLUTIONARY AIR CYLINDERS





THE ONLY AIR CYLINDERS WITH ALL CONTROLS BUILT-IN

HERE are Air Cylinders so different in design that they have become known as the Bellows Air Motors. They are not rotary motors but double-acting, reciprocating, cylinder-type power units. Four-way directional valves and separate speed control valves for advance and retract strokes are built-in. Only one air connection is needed, which can be flexible hose.

Mechanically operated models are equipped with an operating lever which works from any angle in any plane. It may be operated by hand or foot, or may be linked mechanically with any moving element.

Electrically operated models are designed for electrically interlocked equipment.

Bellows Air Motors are available in bores of 1¼", 2½", 3⅝" and 4½", and in any stroke length. They can be mounted in any position.

WRITE FOR FREE BOOKLET

Complete description of Bellows Air Motors, technical data, dimensional tables, case histories, etc. Ask for bulletin BM-20. Address: The Bellows Co., Dept. MD-151, 222 W. Market St., Akron 9, Ohio.

• HYDRAULIC CHECKING OF AIR POWER

The Bellows Hydro-Check removes the natural "bounce" and "springiness" from air — gives the smoothness of hydraulic operation to air-powered equipment — but keeps the speed, flexibility and economy of air-operation. The Hydro-Check is available for use with all Bellows "Controlled-Air-Power" Devices or may be installed to regulate and control the piston movement of standard Air Cylinders.

The Bellows Co.
AKRON, OHIO

VISIT BOOTH 802 ASTE SHOW AND SEE HOW THESE DIFFERENT AIR CYLINDERS CAN HELP YOU LOWER PRODUCTION COSTS AND IMPROVE MACHINE PERFORMANCE

and other mechanical and electronic devices manufactured by Kollsman fit well into the production facilities of Standard Coil. A wholly owned subsidiary of Standard Coil will take over the Kollsman business without change of personnel.

♦

Specially designed materials handling equipment will be a feature of the new plant at Pratt Oval, Glen Cove, N. Y., to which **Steiner Plastic Mfg. Co.** is moving. The move will triple the company's manufacturing space to 75,000 sq. ft. Steiner specializes in the forming and fabrication of thermoplastic sheets, rods and tubes. Present offices will be maintained at Long Island City, N. Y., until the end of 1951.

♦

Sterling Electric Motors Inc. has acquired an 11-acre site in Van Wert, O., for the construction of a branch plant to serve the company's midwestern and eastern business. Plans and specifications for the new building and equipment are being prepared and it is estimated that the investment will be close to one-half million dollars.

♦

Hudson Wire Co., Ossining, N. Y., manufacturer of fine wires, is planning the construction of another magnetic wire plant. The new plant will be located at Cassapolis, Mich., and will supplement the production facilities of Hudson plants at Ossining, N. Y., Winsted and Norwalk, Conn., and Pownal, Vt. The company manufactures bare and insulated wires for electrical and radio-electronic applications, as well as wires and screens for mechanical and chemical needs.

♦

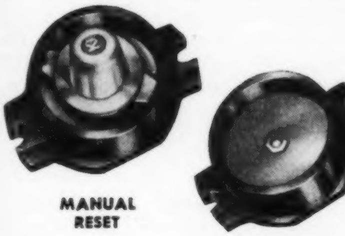
Porter-Cable Machine Co., Syracuse, N. Y., has purchased the Johnson Engineering and Sales Corp., Rockford, Ill. The purpose of the move is to extend the Porter-Cable line of electric tools to include a wide selection of portable woodworking machines.

♦

Ward Leonard Electric Co., Mount Vernon, N. Y., has established an Industrial Chrome Division. This new division is engaged in the development and manufacture of chrome plating units and solutions for industrial hard chrome plating of machine tool accessories. Exclusive license for the manufacture and sale of the chrome plating solution and unit used in the process was obtained from the Chromecraft Corp., Chicago, Ill.

Here's What Service Managers Say About

KLIXON PROTECTORS



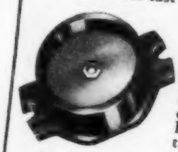
Appliance service managers, electric motor repairmen, heating and air conditioning dealers all over the country find that motor burnouts in electrical appliances of all types are practically eliminated when these motors are equipped with Klixon Protectors.

You, too, can provide your dealers and consumers with motor burnout protection by requesting and specifying that your motor manufacturer supplies you with motors that have Klixon Protectors built in. The additional cost is exceptionally low while the benefits are high. Just take a look at what the service managers have to say and you'll readily see for yourself why it pays to use motors with Klixon Protectors.

Department Store Service Manager Credits Klixon Protectors with Reducing Service Calls

BALTIMORE, MD.: Mr. John Devoto, Appliance Service Manager of Hecht Bros., one of Baltimore's and Washington's leading department stores, praises Klixon Protectors for minimizing service calls.

"We find that appliances using Klixon Motor Protectors have cut out the number of service calls to a bare minimum on motor troubles. I can count the number of times on one hand that we have had to replace a Klixon-protected motor in the last three years."



The Klixon Protector illustrated keeps motors in electrical appliances and other motor driven equipment from overheating and burning out. Look for equipment with Klixon Protected motors for trouble-free motor operation.

KLIXON

SPENCER THERMOSTAT
Div. of Meta... ...trols Corp.
000 FO...
ATTLE...

Appliances Service Mgr. Applauds Klixon for preventing motor burn-outs and keeping customers happy

HADDONFIELD, N. J.: Roger K. Haines of Roger K. Haines Associates, refrigeration and radio, television service company finds Klixon Protectors a big savings.

"Klixon overload protectors have resulted in a big saving in service on both television and refrigeration. On motor-driven appliances we find that Klixon prevents motor burnouts completely, which makes our servicing work easier and keeps our customers very happy."

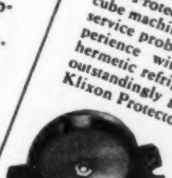
The Klixon Protector illustrated is built into the motor by the motor manufacturer. It keeps motors ... equipment as re... burners, ...nes, etc., preventing ...n burning ...rvice calls, ...rs and re... ...uest equip... motors with ...ors.

...IERMOSTAT
Controls Corp.
...EST STREET
...O, MASS.

Commercial Service Manager Praises Klixon Protection

YORK, PA.: William G. MacBride, Commercial Service Manager of the York Corp., was quick to give credit to Klixon Protectors for reducing their service problems.

"The application of a Klixon Inherent overheat Protector to the solenoid valve in our ice cube machine has unquestionably reduced our service problems on this equipment. Our experience with Klixon Protectors on our hermetic refrigeration compressors has been outstandingly good, also. We think a lot of Klixon Protectors."



The Klixon Protector illustrated keeps motors in electrical appliances and other equipment from overheating and burning out. Look for equipment with Klixon Protected motors.

KLIXON

SPENCER THERMOSTAT
Div. of Metal & Controls Corp.
000 FOREST STREET
ATTLEBORO, MASS.

These advertisements appear in Business Week, Electrical Merchandising, Electrical Dealer, Air Conditioning & Refrigeration News, Domestic Engineering.

KLIXON

SPENCER THERMOSTAT
Division of Metals & Controls Corp.
2501 FOREST ST., ATTLEBORO, MASS.

SOCIETY ACTIVITIES

The **Society of Automotive Engineers** (Detroit Section) has announced a new award for the encouragement of younger engineers. The award is called the Henry Ford Memorial Award in honor of the late automobile manufacturer. It was established by the Detroit Section, any SAE member under 33 years of age being eligible to compete. Competition is through original papers which are presented to or are suitable for presentation to an SAE meeting. Content of competing papers is limited to subjects related to automotive ground vehicles and must describe the work or investigation with which the author has been directly associated. The award consists of a suitable certificate of merit and a cash prize of $200.

♦

American Society of Mechanical Engineers presented the following awards during its 1950 Annual Meeting, Statler Hotel, New York, N. Y.

ASME Medal to Harvey C. Knowles, vice president of the Proctor and Gamble Defense Corp., for his contributions to the application of time and cost reducing continuous production processes to the loading of explosive ammunition.

Holley Medal to Charles G. Curtis, president of the International Curtis Marine Turbine Co., for his inventions of great public penefit. Mr. Curtis is famous for his steam turbine and the first gas turbine power plant in America.

Worcester Reed Warner Medal to Orlan W. Boston, professor and chairman of the Department of Metal Processing, University of Michigan, for his contributions to the art of cutting metals which improved American productive capacity.

Richards Memorial Award to Burgess H. Jennings, professor and chairman of the Department of Mechanical Engineering, Northwestern University, for his contributions to the practice and literature of refrigeration and air conditioning.

Melville Prize Medal to Samuel J. Loring, consulting engineer, Hamilton Standard Propeller Division United Aircraft Corp., for the best original paper on any mechanical engineering subject presented to the society during the previous year. Mr.

Loring's paper is "A Theory of the Mechanical Properties of Hot Plastics."

Pi Tau Sigma Gold Medal to Arthur P. Adamson who is in charge of a group in the Navigation Control Section, Systems Development Division, General Electric Co. This group is concerned with the development of steering systems for guided missiles.

•

American Society of Refrigerating Engineers installed a new president during its Forty-Sixth Annual Meeting in New York City. Paul B. Christensen, vice president and chief engineer of Markets Refrigerating Co., will guide the society's activities for 1951. Other officers elected are: Vice presidents — Edward Simons, San Francisco, and Richard C. Jordan, Minneapolis; treasurer—Donald K. Trescler, Chicago; Members of Council—C. M. Ashley, Syracuse, Leon Buehler Jr., Chicago, Oliver C. Eckel, Boston, Milton Kalischer, Springfield, Mass., D. C. McCoy, Dayton, and J. R. Hornaday, Muskegon, Mich. Election was by letter ballot of the 5600 members.

•

American Standards Association has elected H. S. Sizer to its board of directors. Mr. Sizer, assistant to the director of design, Brown & Sharpe Co., will serve a three-year term commencing January 1, 1951.

•

Society of Naval Architects and Marine Engineers at its Annual Meeting elected J. H. King president of the society for a two-year term commencing January 1, 1951. Mr. King is vice president of the Babcock & Wilcox Tube Co.

•

Lawrence B. Richardson, rear admiral, USN (ret) has been elected president, **Institute of Aeronautical Sciences.** Serving as vice presidents are: Raymond D. Kelly, superintendent of technical development, United Air Lines Inc.; William C. Rockefeller, assistant to the chairman of the board, Consolidated Vultee Aircraft Corp.; William T. Schwendler, executive vice president and director, Grumman Aircraft Engineering Corp.; and Edward C. Wells, vice president of engineering, Boeing Airplane Co. The 1951 treasurer is E. E. Aldrin, aviation manager, Atlas Supply Co. Installation of officers will take place at the Honors Night Dinner held during the Nineteenth Annual Meeting of the IAS at the Hotel Astor, New York, January 29 to February 1, 1951.

SALES AND SERVICE PERSONNEL

FORMERLY assistant manager of V-belt sales, **Wilbur E. Combs** has been appointed product manager for the L. H. Gilmer division of United States Rubber Co. In his new position, Mr. Combs will be responsible for sales of V-belts, shock pads, flat transmission belts and other Gilmer products, making his headquarters at the Gilmer plant in Tacony, Philadelphia. The company has also announced the appointment of **Joseph A. Conlon,** former district sales manager of its Chicago branch, as manager of allied sales for the mechanical goods division. **Edwin D. Meade,** former manager of western railway sales, was appointed to replace Mr. Conlon in Chicago.

•

Don Smith, plant manager of the Peerless plant of Wellman Bronze & Aluminum Co., Cleveland, for the past six years, has been appointed general plant manager. His responsibilities now include managership of the Superior plant at 6017 Superior Ave., as well as the Peerless plant at 2525 East 93rd St.

•

Two sales promotions have been announced by Raybestos-Manhattan Inc., Manhattan Rubber Division, Passaic, N. J. **John T. M. Frey** has been appointed assistant manager of the New York branch and **Lamar S. Hilton** has been named assistant sales manager of the abrasive wheel department.

•

The Hartford Special Machinery Co., Hartford, Conn., has announced the appointments of **Robert A. Bode** as sales manager and **J. James Tasillo** as assistant sales manager for its complete line of automatic drilling and tapping machines, die polishing machines and other products.

•

Richard H. DeMott, who started as a salesman with SKF Industries Inc. thirty-five years ago, is now president of the company. He succeeds **William L. Batt,** wartime production expert, who formally resigned to re-enter government service. Mr. DeMott, who has been vice president in charge of sales since 1943, played an important part in pioneering the use of antifriction bearings in the papermaking, textile, railroad, elec-


Low cost bushings with HIGH performance
Rolled split-type bronze bushings, produced in large volume, in countless varieties, for hundreds of uses, offer *you* economies for your applications. Rigid quality control in production assures the desired performance standards. Controlled alloy quality and grain structure mean uniformity, strength and free machinability. A thin-wall can be specified in a Federal-Mogul bushing without loss of strength, saving material, time and labor. A great range of lengths and diameters is produced and can include oil holes, formed oil grooves, slots, cut-outs, standard or special seams. For heavy-duty, high precision or intricate shapes we also manufacture a wide variety of cast bushings to almost all bronze specifications. Ask your Federal-Mogul representative, or write:
FEDERAL-MOGUL CORPORATION
11045 Shoemaker • Detroit 13, Mich.
Our six plants produce sleeve bearings in all designs and sizes, cast bronze bushings, rolled split-type bushings, bi-metallic rolled bushings, washers, spacer tubes, precision bronze parts and bronze bars.
FEDERAL-MOGUL
Since 1899

tric motor and other industries. SKF has also announced the appointment of **E. A. Erickson** as manager of its Hornell, N. Y., plant and the promotion of **Harry R. Fillmore** from general foreman to assistant plant manager. In addition, **William F. Hagen,** has been appointed assistant district manager of the company's New York office.

•

Philip C. Neumann and **William C. Campbell Jr.,** both electrical engineers, have been named sales representatives in the Pittsburgh district office of Allis-Chalmers Mfg. Co.

•

Recently announced were the promotions of **H. Logan Lawrence, William H. Ayscue** and **Samuel W. McCune III,** to the newly created positions of sales supervisors in the Boston, Chicago and New York district offices, respectively, of the Rubber Chemicals division of E. I. du Pont de Nemours and Co. Inc.

•

Wayne Belden has been elected executive vice president; **Charles Belden** has been made vice president; and **Robert G. Cady** has been named sales manager of the Ajax Flexible Coupling Co. Inc.

•

The appointment of **Arthur W. Eichmann** to its sales engineering staff has been announced by Lear Inc. Mr. Eichmann's experience as head of the electrical test department for General Electric Co. and more recently as procurement engineer for the Glenn L. Martin Co. make him well qualified to render valuable customer assistance in the solution of electromechanical actuating and control systems problems.

•

Price M. Davis Jr., 5807 North Crestwood Blvd., Milwaukee, Wis., has been appointed factory representative by Hose Accessories Co. for Michigan, eastern North Dakota including Bismark, and eastern South Dakota including Pierre. Mr. Davis will also continue to cover Minnesota, Wisconsin, northern Illinois and northern Indiana.

•

The Lincoln Electric Co. recently announced several changes and additions in personnel. **James William Brooks** has assumed sales and engineering responsibilities in the Indianapolis district, having been transferred from the company's Boston district. **Thomas L. Dempsey** has joined the staff of the Cleveland sales organization as a special field engi-

neer. In addition to duties as sales engineer, he will provide engineering service to manufacturers in the Cleveland area who are redesigning machinery for welded steel fabrication. **John F. Kotchian** is now serving industrial accounts for the company as a welding engineer in the Chicago district.

•

M. S. Klinedinst has been named manager of the industrial equipment sales section of the Radio Corporation of America engineering products department, succeeding **P. B. Reed,** who was recently made vice president in charge of the government service division of the RCA Service Co. Mr. Klinedinst, a veteran of 17 years in the engineering and sales activities of RCA, was formerly manager of the scientific and industrial equipment sales section of the RCA International division.

•

Appointment of **M. J. Dunn** as field sales manager of the clock division of Telechron Inc. was announced recently. In his new position, Mr. Dunn will be responsible for all sales activities of the standard clock division.

•

Robert A. Miller has been appointed sales manager of the gear coupling division of Sier-Bath Gear & Pump Co. Inc., 9252 Hudson Blvd., North Bergen, N. J. Formerly assistant sales manager of the coupling division, Mr. Miller is now in charge of appointing representatives for the company's new line of flexible gear couplings.

•

Announcement has been made of the appointment of **O. P. Robinson** as manager of the Pittsburgh district sales office of Cutler-Hammer Inc., replacing **T. S. Towle,** who is now retiring after an association of thirty-six years with Cutler-Hammer. Mr. Robinson joined the company in 1936 as a member of its Chicago district sales office and in 1940 was transferred to Pittsburgh. In his new position he will also supervise the company's Youngstown office in the sale of motor control and allied electrical apparatus.

•

Joseph F. Eckel, manager of the General Electric plant at Lynn, Mass., since 1947, has been appointed manager of the company's large motor and generator divisions at Schenectady, N. Y. He succeeds **J. M. Crawford,** who was recently transferred to Pittsfield, Mass., as manager of the transformer and allied product divisions. **Herbert L. Ross,** former manager of

manufacturing of the meter and instrument divisions, replaces Mr. Eckel at the Lynn plant. General Electric also recently announced three new appointments in its small and medium motor divisions. **J. T. Farrell** has been named assistant to the manager of sales of the small and medium motor divisions, and **Howard W. Bennett** and **Paul D. Ross** have been named managers, respectively, of the new gear-motor and packaged drive sales division and another new group, the Erie, Pa., armored motor sales division.

•

Wilbur R. Leopold has been appointed assistant to vice president **T. Cruthers** of Worthington Pump and Machinery Corp., Harrison, N. J., and will function on public works projects. Replacing Mr. Leopold, **Clarence S. Wentworth** will fill the post of manager of the company's Detroit office.

•

The appointment of **E. J. Waters** as sales manager of its automotive lubrication division was announced by the Aro Equipment Corp., Bryan, O. After an extensive traveling survey of Aro's field representation he will assume his new duties at the Bryan office.

•

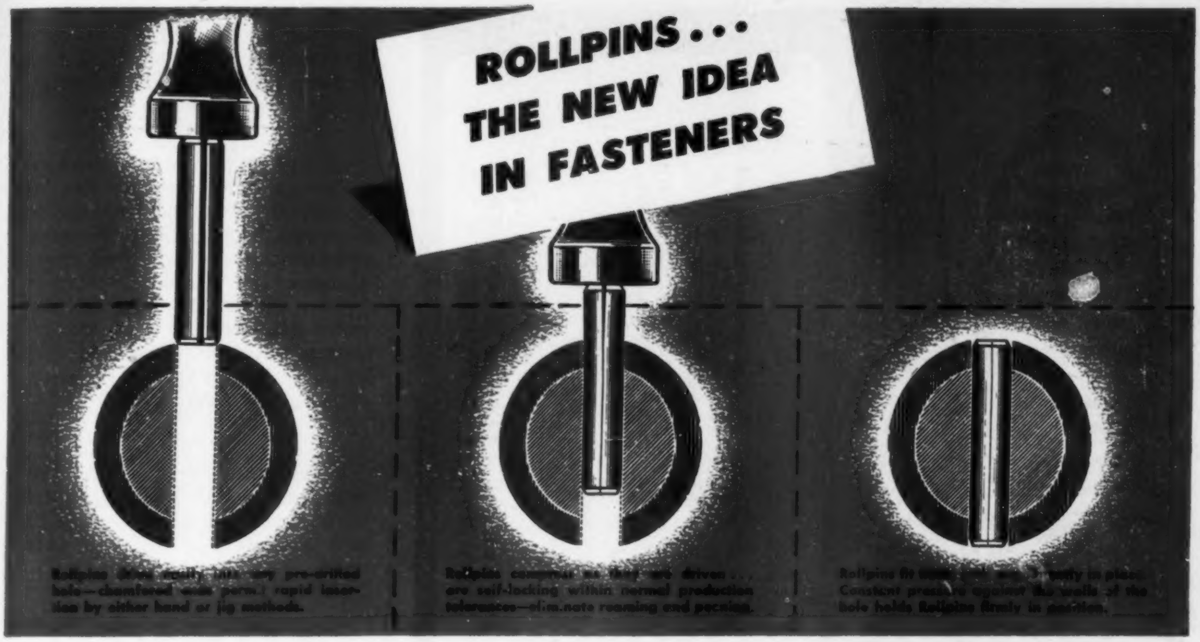
F. T. Harrington, vice president of sales for Vickers Inc., division of the Sperry Corp., 1400 Oakman Blvd., Detroit 32, Mich., recently announced the appointment of **R. E. Esch** as general sales manager. Mr. Esch has been industrial products sales manager since 1946 and has been with Vickers Inc. for a total of seventeen years.

•

Ronald C. Hinman has been appointed to the position of sales engineer with Western Gear Works and has been assigned to the aircraft industry activities of the company. His office will be at the company's Lynwood plant in Los Angeles County.

•

Several new appointments were announced recently by the Carpenter Steel Co., Reading, Pa. **John W. Thompson** has been named product manager of the company and will supervise stainless, alloy and tool steel sales. **Harold R. Potter** is now sales manager for the Cleveland district with headquarters in Cleveland. He succeeds **James S. Bailey,** who has been named assistant to the vice president in charge of sales to devote his time to special work. Also, **William R. Staples,** former West Coast representative, has been ap-



How much can Rollpins save on your production line?

Here's important information on Rollpins—the amazing new fasteners that eliminate slow, expensive reaming, peening, and machining operations. Just imagine the cost-cutting possibilities provided by a single fastener with such wide design and application flexibility that it can replace tapered pins, grooved pins, or straight pins. Investigate the savings Rollpins offer *your* product.

In the short period since their introduction, manufacturers are already using Rollpins as steel fastening pins holding pulleys and gears to shafts; as pivot or hinge pins, clevis pins, cotter keys, shafts, and locating dowels . . . to provide lower-cost, simplified, vibration-proof assemblies.

Rollpins require no special installation skills . . . readily replace your present fastener . . . exceed the sheer strength of a cold-rolled pin of equal diameter. Rollpins stay tightly in place until deliberately removed with a pin punch—can be used over and over again.

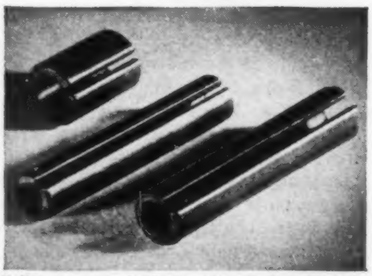
For complete information on Rollpins and their almost unlimited money-saving applications write to Elastic Stop Nut Corporation of America, 2330 Vauxhall Road, Union, New Jersey.

ROLLPIN

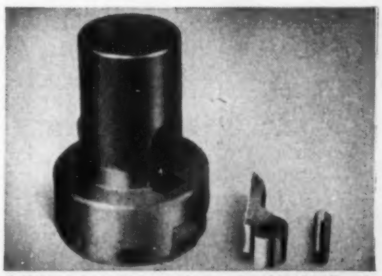
A PRODUCT OF

ELASTIC STOP NUT CORPORATION OF AMERICA

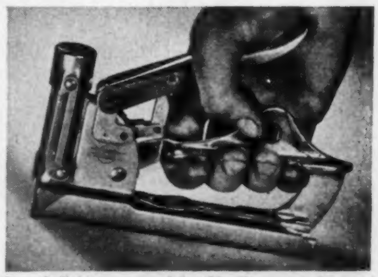
ESNA TRADE MARK



Rollpins are made from either Carbon Steel or Stainless Steel and are readily available from stock in diameters from 1/16 inch to ½ inch and in a broad range of standard lengths.



Rollpins are used to replace a hardened, ground tapered pin in this feed tube finger clutch assembly—stand up to flexing and shock more than 2,400 times an hour.



Four Rollpins are used in this Hansen tacker as pivots. Self-retaining, they eliminate headed rivets and bolts . . . simplify maintenance operations . . . provide a flush fit.

pointed assistant to the manager of sales for the Alloy Tube division of the company in Union, N. J. **Paul E. Kelly** has taken over Mr. Staples' former duties.

♦

The appointment of **Frank G. Drake** as general sales manager has been announced by Bede Products Inc., Cleveland. He succeeds **Richard E. Estabrook,** who was named general manager. A veteran of thirty-five years in the finishing field, Mr. Drake will handle sales for Bede Products, manufacturers of paint heaters for all types of industrial and automotive finishing applications.

♦

Phillips B. Patton, former field engineer, has become manager of the sales engineering department of Lenkurt Electric Co. Inc., San Carlos, Calif.

♦

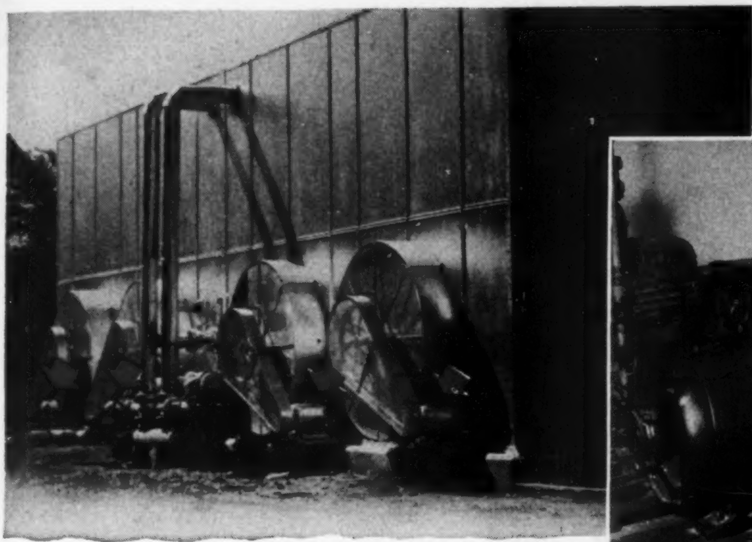
E. C. Barlow has become affiliated with Charles J. Haas Inc., manufacturers of industrial chemicals, oils and greases, with manufacturing plant and offices at American and Cumberland Sts., Philadelphia. Mr. Barlow was with E. F. Houghton & Co. for twenty-four years in the capacity of sales manager for the central division and later served in Philadelphia as assistant to the executive vice president.

♦

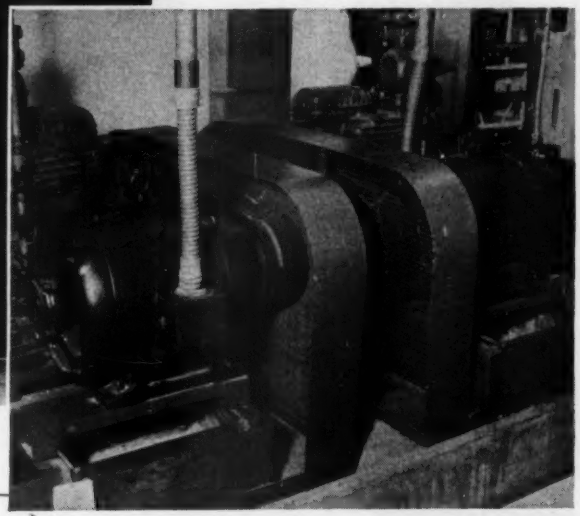
Announcement was made recently of the promotion of **Carl Bauer** to the newly created post of vice president in charge of industrial sales of Standard Varnish Works, New York and Chicago. He was formerly industrial sales manager of the firm's New York operations. His new duties represent a consolidation of the Eastern and Western sales activities under a single administrative head. **J. Peter Jordan,** industrial sales manager of Standard Varnish Works, Chicago, will continue to serve in that capacity under the new consolidated organization.

♦

James J. Filas has been appointed manager of the newly-created fastener department of Acme Steel Co., Chicago. A member of the Acme Steel sales staff for over twenty-five years, Mr. Filas has served in New England and Midwestern territories, and until the new appointment, had been special representative in charge of stitching machine activities in the central division. The fastener department will be responsible for the development of manufactured steel specialty items used for fastening and tacking.



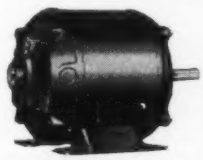
Four Century 5 horsepower splash proof motors driving fans.



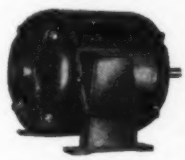
Two Century 30 horsepower motors driving refrigeration compressers.

Century

TYPE SC-
SQUIRREL CAGE MOTORS

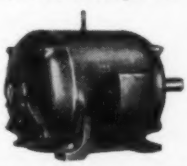


⅛ to ¾ horsepower



1 to 1½ horsepower

2 to 15 horsepower



20 to 125 horsepower

150 to 400 horsepower

Equipment Producers Select Century Motors

To Give You Long-Life Performance With Least Down-Time

The correct selection of the right combination of motor type, speed, power, torque, frame and mounting keeps Century motors on the job.

Team work between your motorized equipment producers and Century motor engineers means that you always get the right motor—selected from Century's wide range of types and kinds, in sizes from ⅛ to 400 horsepower for single or polyphase alternating current and direct current. You can be confident that you get top performance from the fine equipment these motors drive.

Skillful application makes sure that Century motors meet the exacting requirements of the machines they drive. That's your assurance of dependability.

CENTURY ELECTRIC CO. 1806 Pine St., St. Louis 3, Mo.

Offices and Stock Points in Principal Cities

CE-690R

Specify Century C MOTORS for all your electric power requirements

SALES NOTES

To PROVIDE improved service and sales engineering advice, **George K. Garrett Co., Inc.**, Pennsylvania manufacturer of lock washers, flat washers, hose clamps, retainer rings, springs, and stampings, has just moved to its own building at 15755 James Couzens, Detroit 21, Mich. This Detroit branch is under the direction of J. A. Cotter, new district sales manager.

•

According to a recent announcement, **McConkey-Docker & Co.**, 126-132 West Madison St., Phoenix, Ariz., has been appointed an authorized distributor for the entire state of Arizona by **Carboloy Co. Inc.** of Detroit.

•

The **Engineering Products Co.**, Charleston, W. Va., has been appointed sales representative for the full line of Eriez magnetic separation equipment, which includes all permanent magnetic separators made by **Eriez Mfg. Co.** of Erie, Pa., as well as RCA electronic metal detectors. Engineering Products Co. will make available complete research and laboratory testing facilities through the co-operation of the Eriez organization.

•

The appointment of **Donald Sales and Mfg. Co.**, 6601 West State St., Milwaukee 13, Wis., as exclusive distributors in Wisconsin for its complete line of metal cleaners, Bright Copper and other metal finishing preparations, has been announced by **MacDermid Inc.** of Waterbury, Conn. T. F. O'Brein, MacDermid sales engineer, will be available for customer service concerning any metal finishing problems.

•

Ladish Co., Cudahy, Wis., manufacturer of a complete line of forged and seamless welding pipe fittings, has announced the establishment of a branch office at 405-406 Thompson Bldg. in Tulsa, Okla. The office is under the managership of G. E. Mahoney, who for the past three years has served as district manager of Ladish at Chicago. Simultaneously Ladish announced a change in address of its St. Louis office and the appointment of W. H. Heckenberg as

district manager. The new office address is Suite 1605, Continental Bldg., 3615 Olive St., St. Louis, Mo.

•

The **S. J. Smith Co.,** 410 West River St., Davenport, Iowa, has been appointed an authorized dealer of products of the **Air Reduction Sales Co.,** a division of **Air Reduction Co. Inc.** The new company, which opened last November, is owned and operated by S. J. Smith and will service Scott County in Iowa and Rock Island County in Illinois with a complete line of Airco gas and electric arc welding equipment, supplies and accessories.

•

For better service to industries in the greater Pittsburgh area, **Link-Belt Co.** has constructed and is now operating a new factory branch store located at 5020 Centre Ave., Pittsburgh 13, Pa. This new building provides ample space for stocking transmission and materials handling products and serves as headquarters for the increased Pittsburgh personnel.

•

Roots-Connersville Blower Corp., Connersville, Ind., has announced the appointment of the **Koerner Engineering & Supply Co.** of Portland, Ore., as exclusive sales agents for all R-C products in the states of Oregon and Washington. Roots-Connersville manufactures rotary positive and centrifugal blowers and exhausters, cycloidal vacuum pumps, positive displacement meters, and inert gas generators.

•

The following new sales representatives have been appointed by **Double Seal Ring Co.** of Fort Worth, Tex.: **Alton J. Fabrey,** representating the eastern part of the state of New York; **Clifford J. Lane,** in the western part of the state of New York; and the **C. H. Garrison Co.,** covering the state of Kansas and southwestern Missouri.

•

Joseph T. Ryerson & Son Inc., steel distributor, has moved into its new and larger steel-service plant and office building in Cincinnati, construction of which was begun last January. The new plant, located at 3475 Spring Grove Ave., represents an investment of well over $1,000,000 and replaces the company's former plant at Front St. and Freeman Ave. The mail address of the company, Box 300, Cincinnati 14, O., remains unchanged. Complete equipment has been installed for cutting and otherwise preparing steel to customers' requirements.

MEETINGS
AND EXPOSITIONS

Jan. 15-18—

Plant Maintenance Show to be held in the Public Auditorium, Cleveland. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 18-20—

Society of Plastics Engineers. Seventh annual national technical conference to be held at the Hotel Statler, New York. Additional information may be obtained from society headquarters, 409 Security Bank Bldg., Athens, Ohio.

Jan. 22-26—

American Society of Heating and Ventilating Engineers. Tenth international heating and ventilating exposition to be held at the Commercial Museum in Philadelphia, Pa. Additional information may be obtained from the International Exposition Co., 480 Lexington Ave., New York, N. Y. Charles F. Roth is manager.

Jan. 22-26—

American Institute of Electrical Engineers. Winter meeting to be held at the Statler Hotel, New York. H. H. Henline, 33 West 39th St., New York 18, N. Y., is secretary.

Jan. 29-Feb. 1—

Institute of the Aeronautical Sciences. Nineteenth annual meeting to be held at the Hotel Astor, New York. R. R. Dexter, 2 East 64th St., New York 21, N. Y., is secretary.

Feb. 18-22—

American Institute of Mining and Metallurgical Engineers. Annual meeting to be held at the Jefferson Hotel, St. Louis, Mo. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Feb. 28-Mar. 2—

Society of the Plastics Industry. Sixth annual techical session of the Reinforced Plastics Division to be held at the Edgewater Beach Hotel, Chicago, Ill. W. T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

Mar. 5-9—

American Society for Testing Ma-

terials. Spring meeting to be held at the Netherlands-Plaza Hotel, Cincinnati, Ohio. Additional information may be obtained from society headquarters, 1916 Race St., Philadelphia 3, Pa.

March 6-8—

Society of Automotive Engineers. Passenger car, body and materials meeting to be held at the Book-Cadillac Hotel, Detroit, Mich. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Mar. 15-17—

American Society of Tool Engineers. Annual meeting to be held at the Hotel New Yorker, New York, N. Y. Additional information may be obtained from society headquarters, 1666 Penobscot Bldg., Detroit 26, Mich.

Mar. 16—

Institute of the Aeronautical Sciences. Sixth annual flight propulsion meeting to be held at the Hotel Carter, Cleveland, Ohio. R. R. Dexter, 2 East 64th St., New York 21, N. Y., is secretary.

Mar. 19-23—

American Society for Metals. Western metal congress and exposition to be held in the Civic Auditorium, Oakland, Calif. William H. Eisenman, 7301 Euclid Ave., Cleveland 3, Ohio, is national secretary.

Apr. 2-5—

American Society of Mechanical Engineers. Spring meeting to be held at Hotel Atlanta-Biltmore, Atlanta, Ga. C. E. Davies, 29 West 39th St., New York 17, N. Y., is secretary.

Apr. 16 —

Packaging Machinery Manufacturers Institute. Semi-annual meeting to be held at the Hotel Dennis, Atlantic City, N. J. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

Apr. 16-18—

American Society of Lubrication Engineers. National convention to be held at the Bellevue-Stratford Hotel, Philadelphia, Pa. W. F. Leonard, 343 South Dearborn St., Chicago 4, Ill., is secretary.

April 16-18—

Society of Automotive Engineers. Aeronautic and aircraft engine dis-

play meeting to be held at the Statler Hotel, New York. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Apr. 17-20—

American Management Association. The 20th national packaging exposition to be held in the Atlantic City Auditorium, Atlantic City, N. J. Additional information may be obtained from society headquarters, 330 West 42nd St., New York 18, N. Y.

Apr. 23-26—

American Foundrymen's Society. The 55th annual convention to be held in Buffalo, N. Y. Additional information may be obtained from society headquarters, 616 South Michigan Ave., Chicago, Ill.

Apr. 25-26—

Metal Powder Association. Seventh annual meeting to be held at Hotel Cleveland, Cleveland, Ohio. Additional information may be obtained from society headquarters, 420 Lexington Ave., New York 17, N. Y.

Apr. 30-May 4—

Materials Handling Exposition to be held in the International Amphitheatre, Chicago, Ill. Additional information may be obtained from Clapp and Poliak, Inc., 341 Madison Ave., New York 17, N. Y.

Apr. 30-May 11—

British Industries Fair to be held at Olympia and Earls Court, London; and at Castle Bromwich, Birmingham, England. Additional information may be obtained from British Information Services, 30 Rockefeller Plaza, New York 20, N. Y.

May 23-24—

American Society for Quality Control. Fifth annual convention to be held at the Hotel Cleveland, Cleveland, Ohio. Additional information may be obtained from society headquarters, 22 East 40th St., New York 16, N. Y.

May 24-25—

Society of the Plastics Industry. Annual national meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. W. T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

June 3-8—

Society of Automotive Engineers. Summer meeting to be held at the French Lick Springs Hotel, French Lick, Ind. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

How to cut costs when drafting revisions are necessary

* **A case history based on the experience of the Virginia Department of Highways**

1

TODAY the State of Virginia is engaged in a long-range Highway Zoning Program which necessitates changing thousands of drawings to include proposed right of ways.

How to do the job most economically was an important question: Retracing was ruled out—too slow, too expensive. The use of intermediate prints was considered next. They had to be long-lasting . . . easy to make . . . easy to revise.

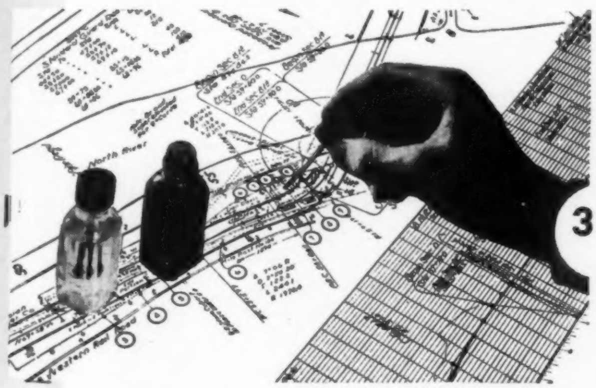
Here's why Kodagraph Autopositive Paper was chosen for the job:

Long-lasting intermediates are assured. In a "permanence test" made by the Virginia Dept. of Highways, an Autopositive print was left on a roof top for 36 days. During this time this photographic intermediate was exposed to 200 hours of sunlight . . . 6.88 inches of rain. Despite all of this abuse it was declared "good as new."

Proof, indeed, that "Autopositives" will stand up under less trying, normal conditions . . . will remain intact in the files year after year . . . ready to produce sharp, clean blueprints whenever needed.

2

Photographic intermediates are produced at a new low cost. When "Autopositive" is used, positive photographie intermediates are produced *directly* without a negative step, without darkroom handling. Maximum efficiency is realized by the Virginia Dept. of Highways because its "Autopositives" are turned out automatically . . . in a continuous blueprint machine, which can be converted readily for Autopositive production.

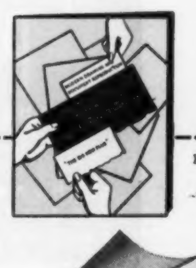


3

Drafting revisions are easily made. Unwanted details—such as existing right of ways—are removed quickly from "Autopositives" with corrector fluid. Then the proposed right of ways are drawn in with pencil or ink. Thus, new masters—prepared without costly redrafting—are ready to turn out the blueprints needed for county supervisors and resident engineers.

Kodagraph Autopositive Paper

"The Big New Plus" in engineering drawing reproduction



MAIL COUPON FOR FREE BOOKLET

- **It enables you, or your local blueprinter, to produce positive photographic intermediates at a new low cost.**
- **It enables you to protect valuable originals from wear and tear . . . obtain more legible final prints.**
- **It gives you photo-lasting file copies.**
- **It restores old, worn drawings . . . reproduces opaques.**

EASTMAN KODAK COMPANY
Industrial Photographic Division, Rochester 4, N. Y.

Gentlemen: Please send me a copy of your illustrated booklet giving all the facts on Kodagraph Autopositive Paper.

Name__________ (please print) Position__________

Company__________

Street__________

City__________ State__________

Kodak
TRADE-MARK

NEW MACHINES

And the Companies Behind Them

Business Equipment

GLUING MACHINE: For applying glue coatings to flexible or rigid materials to ⅜-in. thick and 46 in. wide. Has 6-in. diameter glue and ductor rollers. Glue kept at proper temperature with thermostatically-controlled heating unit in hermetically-sealed tank. Electric variable-speed drive gives speed ratio from 0 to 100 fpm. *Potdevin Machine Co., Brooklyn, N. Y.*

Domestic

IRONERS: Line of automatic flat and rotary ironers. Flat ironer has foot control, features 300-sq in. of ironing surface and 400-lb ironing pressure. Rotary models have controls for knee or hand operations; powered by two-speed fully-enclosed motors. Portable rotary model weighs 33 lb, can be used on any convenient table or stand. *General Electric Co., Bridgeport, Conn.*

REFRIGERATORS: Line of nine new models featuring frozen food compartments extending across the top, defrost indicators, door shelves and butter conditioners. Frozen food capacity in 8.7 cu-ft model, 43 lb; frozen food capacity of 11-cu ft model, 49 lb. Line includes deluxe combination refrigerator-freezers with door shelves, low-cost 6-cu ft model and 4-cu ft undercounter model. *Hotpoint Inc., Chicago, Ill.*

ELECTRIC RANGES: Line includes medium-price pushbutton range, two deluxe pushbutton models, two standard models, and two apartment house models. Various models feature electrically-heated salt conditioner, fluorescent lamp for lighting cooking surface, double ovens, roller-bearing storage drawers, built-in pressure cookers, and timers. *General Electric Co., Bridgeport, Conn.*

AUTOMATIC DISHWASHER: Front-opening, top-loading, fully automatic dishwasher washes and dries 100 pieces of china, glassware and cutlery in 30 minutes. Fits under any standard kitchen counter. Has preliminary power rinse to preheat dishes and wash off loose food particles. Dishes then washed in detergent solution heated by electric heating unit in tub and double rinsed before being dried by hot

air circulated past heating element by fan. *General Electric Co., Bridgeport, Conn.*

ELECTRIC WATER HEATERS: Element ratings changeable to 600, 650, 750, 1000, 1500, 2000, 2500, or 3000 watts. Corox multi wattage heating element is sickle-shaped, has four terminal screws. Changing or omitting jumpers changes wattage. *Westinghouse Electric Corp., Pittsburgh, Pa.*

Heating and Ventilating

DUST COLLECTOR: Model 8N50, rated 885 cfm with 3-in. static suction, recommended where outside exhaust of clean air is required. Requires 20 by 30-in. floor space. Self-clearing, paddle-wheel fan direct-driven from ¾-hp continuous-duty motor, is capable of clearing lint and strings from buffing, chips, shavings and saw dust from woodworking operations, etc. *Aget-Detroit Co., Ann Arbor, Mich.*

GARBAGE DISPOSAL: Can be connected to existing plumbing with estimated 20 per cent reduction in installation costs. Simple locking arrangement permits rotation of unit to line up with existing plumbing. Two inches shorter than previous models, new models have two-quart capacity. One model for installation in existing sinks or new automatic dishwasher sinks, other model for pre-plumbed dishwasher models. *Hotpoint Inc., Chicago, Ill.*

BATHROOM HEATER: Multiple-use unit serves as heater, cooling fan, hair dryer and clothes dryer. Hangs on bathroom wall, pulls out for use as hair dryer. Pull out rack used for hanging clothes for drying. *Fresh'nd-Aire Co., Chicago, Ill.*

Manufacturing

UTILITY GRINDER: New 6-in. deluxe model for tool rooms, machine shops, schools, home workshops, etc. Includes dynamically balanced rotors, integrally cast wheel guards, adjustable tool rests, rubber-mounted cast iron base. Powered by ¼-hp, 3450-rpm, dustproof motor. *The Lima Electric Motor Co., Lima, O.*

SPECIAL DRILLING MACHINE: Drills 7 holes simultaneously in hex-head bolts. Uses 7 Simplex automatic drill units. Bolts loaded manually, held by air clamp. Holding fixture has bushings to handle hex-head sizes from ⅜ to 1⅛ in. Drill units made in sizes from $\frac{1}{16}$ to ⅜-in. drill diameter. *Simplex Tool Engineering Co., Detroit, Mich.*

TOOL GRINDER AND LAPPER: Agathon type 174A precision grinder holds tool in compound holder which slides

on bar parallel to wheel spindle to transfer to lapping wheel. Tools to 1.57 by 2.36 in. can be handled. Special attachments permit grinding and lapping of cutter heads, engraving tools, round boring tools, and other special tools. *Hauser Machine Tool Corp., Manhasset, N. Y.*

PRESS: Capacity, 50 ton. Oil-hydraulic unit has 15-in. stroke, 24-in. daylight opening, 19½ by 31-in. work surface. Approach of ram to work variable, can be preset at any speed to 290 in. per minute with pressing speeds up to 145 in. per minute. Approach and pressing speeds independently adjustable. Ram stroke, length and pressure can also be preset. Either pressure or distance reversal-control included. Optional equipment includes 33-in. diameter hydraulic indexing table. *The Denison Engineering Co., Columbus, O.*

SPECIAL DRILL: For drilling lightening holes in large tractor crankshafts. Handles parts 56 to 73 in. long. Workpiece clamped by manually-controlled hydraulic mechanism, indexed through six stations at each of which two 1½-in. holes are drilled next to one of throws. Can use drills to 2 in. Drills driven at 50 fpm, with minimum infeed of 1 in. per minute. *Snyder Tool & Engineering Co., Detroit, Mich.*

CONTOUR GRINDER: Precision belt grinder attaches to drill press column, swings out of way when not in use. Uses standard 2½ by 60-in. belts, change of belt ratio or spindle speed provides range of 7000 to 36,000 rpm (700 to 4500 fpm). Interchangeable spindle attachment provides continuous roll form or cylindrical grinding. Felt wheel blanks shaped to meet varying requirements. *George F. Grant Co. Inc., West Newton, Mass.*

BANDSAW: Throat capacity, 36 in., work height from 15½ in. up. Uses standard saw blades to 1-in. width, uses automatic synchronized, hydraulic, aircraft type brakes. Powered by 10-hp motor with 3-speed transmission giving tool speed range of 40 to 10,000 fpm. Hydraulically-operated, 40 by 48-in. table supports one ton or more. Table slides on 28 rollers, has 36-in. stroke with 200 lb feeding pressure available. Correct feed pressure automatically controlled by resistance of work. *The DoAll Co., Des Plaines, Ill.*

UNIVERSAL HYDRAULIC GRINDER: Wheelhead swivels 180 degrees, mounts both internal and external spindles. Handles small, large and deep-hole internal work; straight shaft, long shaft, diameter, and shoulder external grinding. Fea-

tures internal spindle speeds from 6000 to 35,000 rpm, all flange mounted spindles, sine bar for setting workhead on table swivel, double swivels on cross slide for two-angle internal and external grinding. *Rivett Lathe & Grinder Inc., Boston, Mass.*

SPOT WELDER: Pedestal rocker arm welder for general use. Rated 5-kva, 220-v, 60-cycle at 50 per cent duty cycle, can make welds in two sheets from 28 to 20-gage steel. Throat depth variable from 3 to 15 in. arm spacing adjusted by swiveling arms. Available with air or water cooled point holders. *Universal Welder Corp., Cleveland, O.*

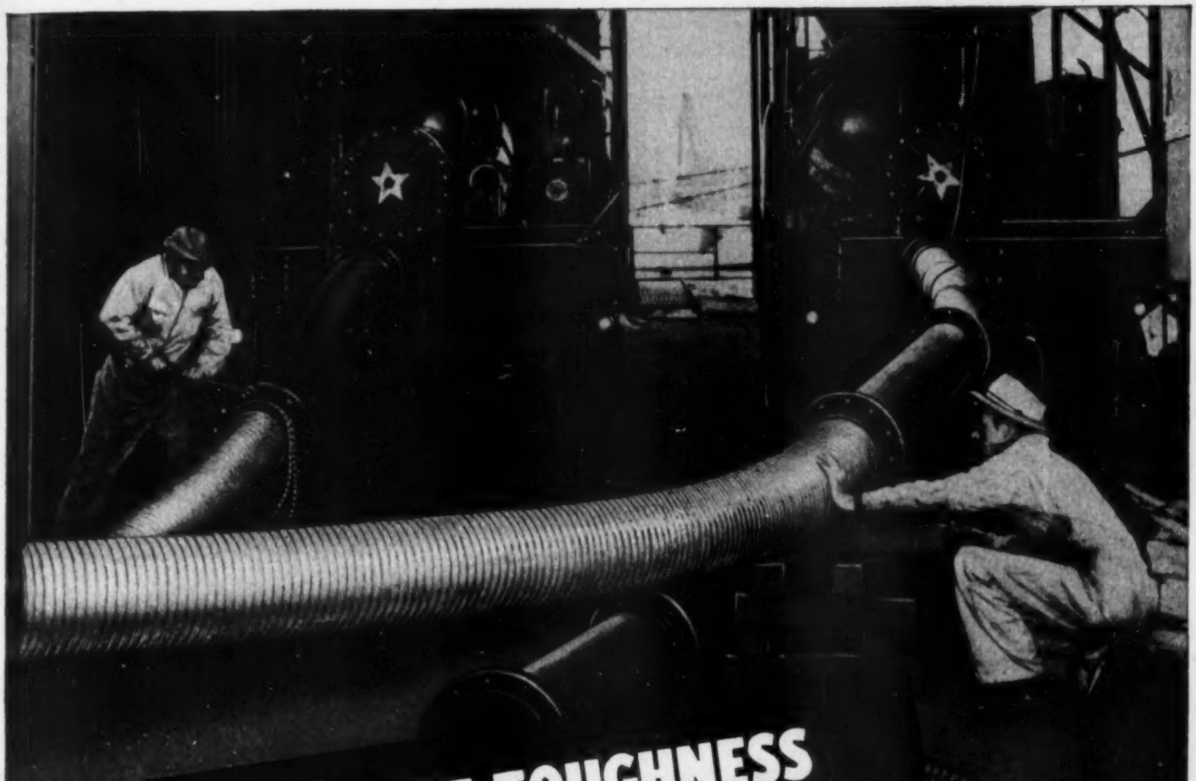
CUTTER AND TOOL GRINDER: Motor-driven power-feed machine will swing work to 12-in., diameter, takes 28 in. between centers, has longitudinal table movement of 24-in. Table size, 6 by 42 in. with single T-slot running length of table. Infinitely-adjustable power-driven longitudinal table speeds between 4 and 100 in. per minute. *Gallmeyer and Livingston Co., Grand Rapids, Mich.*

DIECASTING MACHINE: Capacity, 1 lb. Fully automatic high-speed machine adjustable between 0 to 3000 shots per hour. Cycles entirely automatic, or can be set to stop after one cycle. Opening and closing speeds adjustable. *DCMT Sales Corp., New York, N. Y.*

TUBE BENDING PRESS: Hydraulically operated at speeds to 20 strokes per minute. Positive index stop for various degree bends. Knee width of 11 in. permits reverse bends 5½ in. apart. Overhanging ram permits bending over top of punch. *Gibbons Machine Co., Tipp City, O.*

PORTABLE ARC WELDER: Model 125A for installation, construction and maintenance work. Has 16 heat stages from 20 to 125 amp, handles from $\frac{1}{16}$-in. to $\frac{5}{32}$-in. welding rod, welds iron, steel, brass, bronze and other metals. Operating features include positive control with no arc blow, no moving parts, low initial and operating cost. *Trindl Products Ltd., Chicago, Ill.*

RESISTANCE SPOT WELDER: For spot welding aluminum, magnesium, stainless steel, Inconel, Monel, brass, mild steel and other alloys. Machine converted to welding of any of these materials, and various thicknesses, by change of control settings. Press type machine rated 100 kva at 50 per cent duty cycle; standard throat depth, 36 in., other depths available. Welding range, 0.022 to 0.156-in. on low carbon steel (two thicknesses), 0.025 to 0.081-in. on aluminum and mag-



nesium alloys. Welding current controlled by continuous wave altered to suit thickest or thinnest metals in welder's range. *Sciaky Bros., Chicago, Ill.*

HYDRAULIC PRESS: Capacity, 5000 ton. Originally designed for cold rubber-pad forming of sheet metal parts in airplane frame construction; now used to hot-forge tire retainer rings, axles and other parts for earth-moving equipment. Single-action press has pushbutton and sensitive manual control, is of four-column construction arranged for rapid advance to work. Quick-reversal feature makes operating cycle desirable for hot forging work. *Birdsboro Steel Foundry and Machine Co., Birdsboro, Pa.*

CORE GRINDER: Dual type grinder has column, grinding wheel and drive shaft mounted on large, dust-proof ball bearings. Employs 42-in. wheel with 38-in. effective cutting diameter. Adjustable wheel arm swings in complete 360-degree arc with wheel height above table ranging from 15 to 42 in. Powered by 5-hp, 900-rpm motor driving wheel at 242 rpm. *Milwaukee Foundry Equipment Div., Cleveland, O.*

FLASH WELDER CONTROL: Designed for conversion of mechanically-driven flash welders to hydraulic operation. For welders having transformer capacity to 500 kva. Fully hydraulic unit has calibrated adjustments permitting duplication of set-ups. Enables welding 40 to 1 range of cross sectional areas, within physical and electrical capacities of welder. *Kingsley A. Doutt, Detroit, Mich.*

UNIVERSAL DRESSER: Capable of dressing complete form rather than single radius and angle tangent, made possible through movements in column and base. Angles can be set with sine bare in addition to graduations. When goose neck is removed, may be used as revolving fixture for rolling of radii. All settings contain 0.200-in. step for settings starting from 0.0001-in. and up. *Universal Form Tool Co., Detroit, Mich.*

TRANSMISSION CASE MACHINE: Automatically drills, chamfers, reams and taps 32 holes in ends, sides and tops of 85 automatic transmissioin housings per hour at 100 per cent efficiency. Operations on Transfer-matic machine conducted at 28 stations, parts moving automatically from one station to the next. Machine requires one unskilled operator. *Cross Co., Detroit, Mich.*

SPRING COILERS: Hand-operated machine for making compression, ten-

sion and torsion springs in sample lots or in production runs to 500 pieces. Set up time usually less than 5 minutes, capacity, 300 springs per hour. Compression springs coiled with both ends closed or squared, to any pitch, and with up to 28 coils. Extension springs coiled tight or loose, with up to 100 coils. Torsion springs wound right or left hand with extended arms to 3 in. on both ends. Max OD, ⅞-in.; max overall lengths, 4 in., using wire diameters from 0.005 to 0.065-in. Includes 29 arbors, 3 coiling points and wire cutter. Weight, 45 lb. *The Carlson Co., New York, N. Y.*

Materials Handling

BATTERY-OPERATED TRACTOR: Unit measures 44 in. long. Can be operated from any of three positions; operator facing away from load, facing load, or riding on step. Capacity, 3000 and 5000 lb; drawbar pull, 500 to 800 lb; no-load speed, 3.5 to 4 mph; full-load speed, 2.5 to 3 mph; overall width, 30 in. *Market Forge Co., Everett, Mass.*

DIE HANDLER: For safe handling and inspection of dies. Handles dies to 24 in. wide and 43 in. long, shut height from 9 to 14 in., weighing to 2000 lb. Upper platen has 20 by 42-in. area, has vertical travel of 16½ in. Platen mounted on steel trunions, rotates 360 degrees by hand crank. Locking pin provided at 5 positions. *Morley Machinery Corp., Rochester, N. Y.*

Plant Equipment

HAMMER DRILLS: Electromagnetic Model 10-RO for high-speed drilling in concrete, brick and stone. Automatic rotation of carbide-tipped spiral drill accomplished by rubber ratchet mechanism actuated by each recoil blow. Blows per minute, 3600; weight, 11¾ lb. *Syntron Co., Homer City, Pa.*

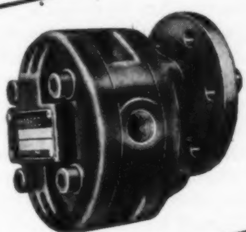
INDUSTRIAL FLUID COOLERS: Accurately control temperature of coolants such as honing, grinding and cutting oils, water, hydraulic oils and plating solutions used in industrial processes. Use air-cooled refrigeration cycle. Coils made from steel or copper tubing. Standard models handle most capacity and temperature requirements, special models available in explosion-proof construction. *Frostrode Div., Viking Products, Vicksburg, Mich.*

ROTARY BAG SEALER: Little Giant bag sealing machine has automatic control of pressure, time and heat. Swivel-mounted head adjustable for sealing bags at any angle from vertical to horizontal. Can be fed

OIL-HYDRAULIC PUMPS • VALVES • CYLINDERS

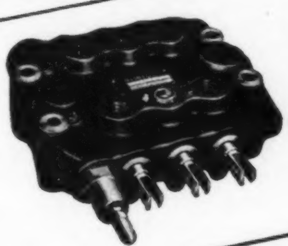
Look to HYDRECO *for* ALL THREE!

Four-Bolt pumps



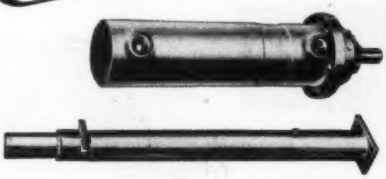
HYDRECO gear type pumps are made in five basic sizes with a variety of flange or base mounted types, ½ GPM to 130 GPM, and for operating pressures up to 1500 p.s.i.

control valves



HYDRECO control valves are made in a variety of sizes and plunger arrangements, in capacities from ½ GPM to 150 GPM, and for operating pressures up to 1500 p.s.i.

cylinder assemblies



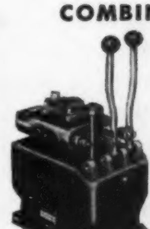
HYDRECO cylinders are made in single-acting, double-acting and telescopic designs in sizes up to 8″ effective diameter. Plain, eye-type, or ends of special design are available for all sizes.

The HYDRECO Line Also Includes:–

AUXILIARY VALVES

Relief, selector, check, cycling, unloading and sequence valves, also pressure and flow regulators and flow dividers.

COMBINATION UNITS



In a wide range of capacities

PVT—Combination valve and tank units with built-in pump.

TV —Combinations of valves and reservoirs in various sizes.

PV —Pump and valve units for special applications.

Send for Bulletin 101 showing complete line of gear pumps, control valves, auxiliary valves, cylinder assemblies and combination pump-valve-tank units.

Write to

HYDRAULIC EQUIPMENT COMPANY
1106 EAST 222ND STREET • CLEVELAND 17, OHIO

HYDRECO
HYDRAULIC CONTROL DEVICES
PUMPS·CYLINDERS·VALVES

THE MARK OF QUALITY

from right or left to suit feeding operation by reversing direction of motor rotation. Equipped with fold-over blade for folding bag tops. *Amsco Packaging Machinery Inc., Long Island City, N. Y.*

GAS GENERATOR: Monogas generator for protective atmosphere work. Available in sizes ranging from 500 to 15,000 cfh. Produces non-decarburizing neutral atmosphere consisting essentially of nitrogen with low hydrogen and carbon content, with carbon dioxide, oxygen and water removed. Includes generator and scrubber in one unit. *Westinghouse Electric Corp., Pittsburgh, Pa.*

PAINT HEATER: Supplies 24 to 28 gph of paint for installations using two or more spray guns. Heats paint to 160-200 F within 15 minutes. Consists of two heating units mounted on common base with circulating arrangement. Paint drawn directly from original container without use of pressure tank. Paint temperature and pressure accurately controlled. *Bede Products Inc., Cleveland, O.*

GAS COMPRESSOR: For compressing natural gas or for use in liquefied petroleum industry. Powered by Lorain Model A multifuel engine, with single-stage compressor driven through overhung crankshaft giving 9-in. compressor stroke from 13-in. engine stroke. Compressor cylinder has bottom discharge to prevent accumulation of gas condensates, is equipped with force-feed lubrication. *White-Roth Machine Corp., Lorain, O.*

CATALYTIC EXHAUST: Renders exhaust gases nonpoisonous and odorless. Adaptable to any type lift or fork truck in place of standard muffler. Weight, 25 to 30 lb. Imposes no additional back pressure on engine; catalyst should be replaced after 3000 hrs operation for max efficiency. Operates only on unleaded gas. *Oxy-Catalyst Manufacturing Co. Inc., Wayne, Pa.*

SPRAY GUN PUMPS: Air-operated, high-volume materials handling pumps for supplying industrial materials through hoses for spray gun, pole gun or extrusion gun application. Pumps operate on pressures from 20 to 175 psi, weigh 35 lb. Include Evenflo device to eliminate spurting. *Gray Co. Inc., Graco Square, Minneapolis, Minn.*

Processing

TUMBLING MACHINE: Increased capacity for grinding, deburring and finishing metal parts. Requires 64

by 68-in. floor area; two 24 by 40-in. ID compartments handle 30 per cent larger work load than comparable machines occupying same floor space. Compartments furnished with ½-in. plate unlined or ¼-in. plate rubber lined. Magnetic brake included on 220/440-v, 5-hp motor. *Grav-i-Flo Corp., Sturgis, Mich.*

BREAD MIXER: Horizontal, high-speed dough mixing and kneading machine. Redesigned model includes addition of access doors to front and sides for easier maintenance and increased sanitation. Double-end agitator drive reduces vibration in machine. Stainless steel clad bowl sheets standard equipment, stainless agitator equipment optional. Agitator consists of three freely rotating bars that knead and mix dough. *American Machine & Foundry Co., New York, N. Y.*

IMPREGNATING EQUIPMENT: For sealing pressure castings rejected because of porosity. Seals all ferrous and nonferrous metals before or after machining. After part has been washed in cold water, no evidence of impregnation remains except that it is pressure tight. Sealant is noninflammable, nontoxic, will not harm the skin. For small parts and castings. *Tincher Products Co., Sycamore, Ill.*

PLATING EQUIPMENT: For economical hard-chrome plating of industrial parts or surfaces up to 10 sq in. at recommended current density of 2 amp per sq in. Compact, portable unit powered by dry disk, power-pack selinium rectifier, is complete with plating bath tank, rheostat, timer, ammeter, and reversing switch for stripping action. Hard chrome deposition can be controlled to tolerances of less than 0.0001-in. *Ward Leonard Electric Co., Industrial Chrome Div., Mount Vernon, N. Y.*

Testing and Inspection

DUCTILITY TESTING MACHINE: To detect surface and subsurface flaws in deep-drawing steel over comparatively large area. Employs 5-in. diameter penetrator. Machine is motorized, hydraulically operated, provides separate controls for clamping and penetrating pressures. Penetrating pressure, to 150,000 lb; clamping pressure, to 100,000 lb. Equipped with set of three dies for testing materials to ¼-in. thick. Short ⅞-in. diameter penetrator on top of large penetrator provides comparison with standard ductility tests. *Steel City Testing Machines Inc., Detroit, Mich.*